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A comparison of various sections of flat-bottomed rail and resultant formulæ,

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GENERAL REMARKS.

Aim and scope of the article.

If a classification is made of various sections of flat-bottomed, or bull-head, rail in order of increasing *weight* (per lineal metre) or of *height* for the purpose of compiling a numerical table or a graph, and taking for the latter purpose the weight or height as abscissa and the geometric or dynamic dimensions as ordinates, it will be seen that the components vary considerably in relation to the same weight B in kgr./m., or to the same height *h*. In the two graphs, broken lines are obtained which have a fairly pronounced slope. All the tables and corresponding graphs in this article clearly show the results stated. not only for rails of the same weight or height but also — and with greater reason — for rails of approximately the same width or height.

This article is addressed to permanent way engineers and metallurgists so as to assist, as far as possible, in the deletion of these saw-toothed lines and their replacement by average lines, straight or curved : only then shall we cease to have tall and thin, or short and thick sections, or sections distorted as regards the proportions of the three parts, base, web and head in sizes and relation to their masses.

The section of a rail should give a moment of resistance to bending, with a well-balanced composition, in accordance with technical, economic and metallurgical demands.

The contents of the article are presented as follows:—

In *Chapter I* are compared 27 rail sections of 30 to 55 kgr./m., (60 to 110 lb. per yard) the elements of which are shown in « Cours de chemin de fer de l'E. N. P. C. », by Profs. Descubes and Martinet. This Chapter covers Tables 1 and 1′, Graphs 2 and 2′ and Table 2_a.

In *Chapter II* are compared:

A. Seven standard French sections, S_1 to S_7, of 20.86 to 55.2 kgr./m. (42 to 110.4 lbs per yard). This part contains diagrams of 2 sections, S_5 and S_6, 4 tables, 5, 5′; 6_b, $6'_b$; 7 and 8, 4 corresponding Graphs 6, 6_a, 7_a, 8_a, a summary of dynamic formulae and a graphic Table 9.

B. 8 Greek State Railways rail sections of 30 to 43.85 kgr./m. (60 to 88.35 lbs per yard). This part includes Table 10 and Graph 10_a.

C. Investigation, and conclusions, concerning the principles governing the constitution of a rail section.

Chapter III gives the linear formulae applicable to 12 German rail sections of 4 to 50 kgr./m. (8 to 100.75 lbs per yard), with Graph 11 and Tables 12 and 13.

Chapter IV deals with the formulae used for calculation of rail resistance and the weight P.

Chapter I.

Details of 27 rail sections, arranged in order of weight, from 30 to 55 kgr./m.

1. *Tables 1 and 1′.*

Corresponding Graphs 2 and 2′.

Taking the letters h, p, a and c to represent the height h of the section, the width p and c of the base and the head, and the thickness a of the web; the same letters accented to represent the heights of the same parts; s as the area in cm^2 of the section; $I : v_1$ the moments of resistance in cm^3 (regardless of I or v_1) and calculating the ratio $p : h$, $p' : h$, $a' : h$, $c' : h$, $c' : c$, $a : a'$, Tables 1 and 1′ and Graphs 2 and 2′ have been formulated showing the zig-zag lines and their general directional lines, with the weights B in kgr./m. as abscissae.

A study of Tables 1 and 1′, for example lines 15, 16, 17 and 18, relating to 46 kgr./m. (92.69 lbs per yard) shows that lines relating to the same weight present a great variety in the components of the same name included in the various lines of the columns of the Table; in Graphs 2 and 2′ (taking the abscissa of 46 kgr./m.) quite different ordinates are shown for the sections in question, the zig-zag lines of each number in Tables 1 and 1′ will be observed, as well as the proposed average lines. Each peak in the broken line coming above or below the average line indicates that the corresponding element is excessive or deficient; for example, the 42 kgr./m. (84.6 lbs per yard) section is excessive as regards p', c', a, p, h, and deficient in a' and c (see Graph 2); the same section is excessive in the ratios $p' : h$, $a : a'$, $c' : h$, $p : h$ and $c' : c$, and deficient in $a' : h$ (see Graph 2′).

TABLE 1

Heights h of rail, c' of head, a' of web, p' of base, functions of weight B, kgr./m.

	B kgr /m.	h mm.	c' mm.	a' mm.	p' mm.	$c' : h$	$a' : h$	$p' : h$
1	29.76	109.54·	35.69	55.56	18.29·	0.326	0.507	0.167
2	30	120	40	57	23	0.333	0.475	0.192
3	30	125	41.25	60.25	23.5	0.330	0.482	0.188
4	32.24	112.71	32.55·	60.32	19.84	0.289	0.535	0.176
5	36	128	39.95	68	20.05	0.312	0.531	0.157
6	36	130	44	58	28	0.339	0.446	0.215
7	37.2	122	40	61.50	20.50	0.328	0.504	0.168
8	38	130	43	63	24	0.330	0.485	0.185
9	39	130	44	58	28	0.339	0.446	0.215
10	39.68	127	38.10	66.67	22.23	0.300	0.525	0.175
11	42	140	47	65	28	0.336	0.464	0.200
12	42	145	57v	53·	35	0.393v	0.366·	0.241
13	45	144	50	64	30	0.347	0.445	0.208
14	45	144	43.5	76	24.5	0.302	0.528	0.170
15	46	141	54.5	57	29.5	0.386	0.404	0.210
16	46	145	48.2	75	21.8	0.333	0.517	0.150·
17	46	145	45	75	25	0.310	0.518	0.172
18	46	152	53	61	38v	0.349	0.401	0.250v
19	47	144	43.5	76	24.5	0.302	0.528	0.170
20	48	141	55	58	28	0.390	0.411	0.199
21	49.6	146.05	46.27	76.2	23.58	0.317	0.522	0.161
22	49.6	146.05	43.26	78.18	24.61	0.296	0.534v	0.170
23	50	149	47	75	27	0.315	0.504	0.181
24	50	144	50.91	63.18	29.91	0.354	0.438	0.208
25	50	150	43	79v	28	0.287·	0.526	0.187
26	52	145	49	72.50	23.50	0.338	0.500	0.162
27	55	155v	53.20	75	26.80	0.343	0.484	0.173
	B kgr /m.	h mm.	c' mm.	a' mm.	p' mm.	$c' : h$	$a' : h$	$p' : h$

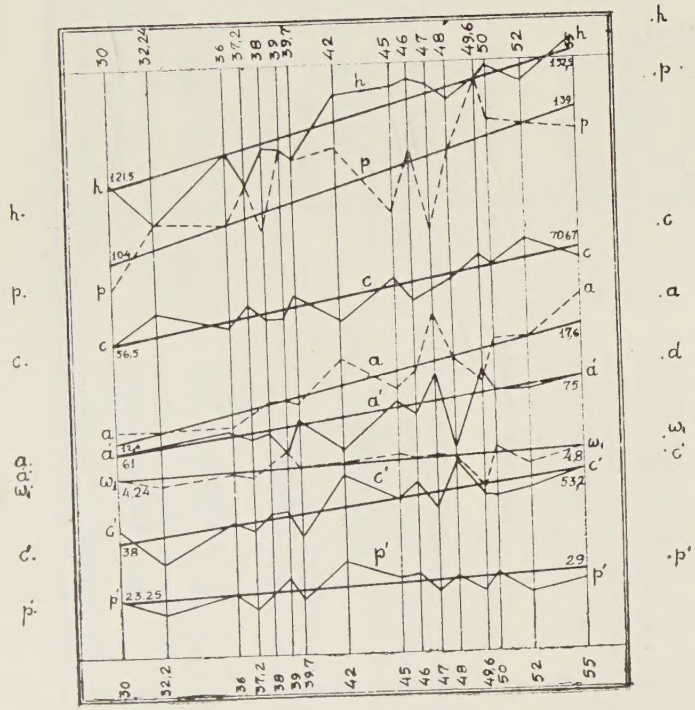
TABLE 1′

Width c of head, thickness a of web, width p of base, area s cm² $\frac{I}{v_1}$ moment of resistance in cm³.

	c mm.	$c' : c$	a mm.	$a : a'$	p mm.	$p : h$	s cm²	$I : v_1$ cm³
1	57.15	0.623	11.51·	0.207	109.54	1.000v	38.69	114
2	57	0.702	13.50	0.237	99	0.825	39.32	125.11
3	56·	0.737	12	0.200	97	0.775	38.71	129.26
4	61.09	0.533·	12.70	0.210	112.10	0.768	41.73	130
5	58	0.689	13	0.191	115	0.898	46.27	153.70
6	60	0.733	13	0.224	110	0.769	45.80	152.40
7	62	0.645	13.50	0.220	122	1.000	47.53	157
8	60	0.694	14	0.222	110	0.846	48.55	157
9	60	0.733	14	0.241	130	1.000v	49.97	188
10	63.50	0.600	13.89	0.208	127	1.000v	50.87	178
11	60	0.783	14	0.215	130	0.928	53.67	191.75
12	60	0.928v	18	0.340v	60·		53.67	190.20
13	60	0.833	15	0.234	120	0,833	57.67	204.84
14	72v	0.604	14	0.184	110	0.764·	57.39	216.80
15	60	0.908	16	0.281	130	0.922	59	202.23
16	62	0.777	15	0.200	134	0.931	59	218.34
17	65	0.692	13	0.173·	125	0.862	58.51	215.48
18	62	0.855	18	0.300	70		58.51	219.26
19	72v	0.604	18	0.237	110	0.764	60.24	220.30
20	66	0.833	16	0.276	130	0.915	61.60	223.20
21	69,85	0.662	14.68	0.193	146v	1.000v	63.65	194
22	69,85	0.619	14.99	0.192	146v	1.000v	63.65	245.20
23	65	0.723	16	0.213	135	0.951	64.79	236.40
24	66	0.771	19.5v	0.308·	134	0.931	64.79	248.50
25	72v	0.597	15	0.190	140	0.933	64.79	253
26	72v	0.681	17	0.234	135	0.931	67.38	232
27	62	0.855	19	0.253	134	0.865	70.75	264.07
	c mm.	$c' : c$	a mm.	$a : a'$	p mm.	$p : h$	s cm²	$I : v_1$ cm³

GRAPH 2

Data of Tables 1 and 1′ relating to 27 rail sections of 30 to 55 kgr./m.



GRAPH 2′

Continuation of Graph 2, relating to 27 rail sections of 30 to 55 kgr./m.

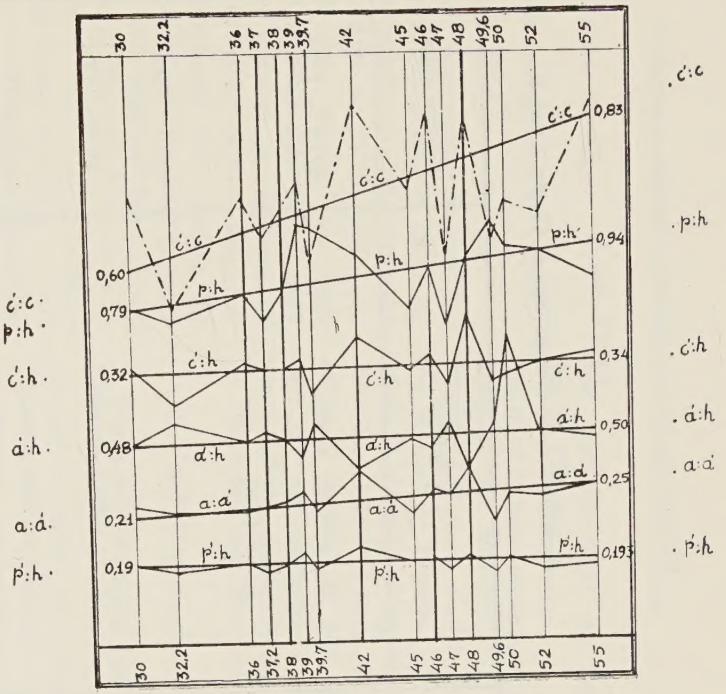


TABLE No. 2*a*

Formulae for the average lines of Graphs 2 and 2′.

Nos.	Formulae	Extreme values	For ± 1 kgr.
1	$h = 1.20B + 84.3$	121.5 — 152.5	± 1.24
2	$p = 1.40B + 62$	104 — 139	± 1.40
3	$c = 0.567B + 39.5$	56.5 — 70.65	± 0.567
4	$a = 0.208B + 6.16$	12.40 — 17.6	± 0.208
5	$a' = 0.560B + 44.2$	61 — 75	± 0.56
6	$c' = 0.608B + 19.76$	38 — 53.2	± 0.608
7	$p' = 0.230B + 16.35$	23.25 — 29	± 0.23
8	$\frac{c'}{c} = 0.00945B + 0.314$	0.6 — 0.83	± 0.0092
9	$\frac{p}{h} = 0.00595B + 0.614$	0.79 — 0.94	± 0.006
10	$\frac{c'}{h} = 0.0006786B + 0.340$	0.32 — 0.34	± 0.0008
11	$\frac{a'}{h} = 0.0009773B + 0.442$	0.48 — 0.50	± 0.0008
12	$\frac{a}{a'} = 0.001886B + 0.149$	0.21 — 0.25	± 0.0016
13	$\frac{p'}{h} = 0.0001B + 0.187$	0.19 — 0.193	± 0.000012
14	$w_1 = 0.0224B + 3.568$	4.24 — 4.84	± 0.024

2. *Table No. 2_a.*

Formulae for the average lines of Graphs 2 and 2'.

The analytical expressions proposed for the average lines are included in Table 2_a, as well as the extreme values proposed for B of 30 to 55 kgr./m. (60 to 110 lbs per yard). In the last column has been noted the plus or minus difference in each value for an increase or decrease of 1 kgr./m. (2.015 lbs per yard) of weight B. It may be mentioned that totalling the figures of items 5, 6 and 7, which give the heights of the web, head and base, should give the figure of No. (1), $h = 1.2\ B + 84.3$ instead of $p' + a' = c' = 1.398\ B + 80.31$. The excess of $0.16\ B - 4$ of the latter total is due mainly to the average ordinates adopted for sections of the same weights, in view of the values of p', a', c'. This difference, equal to o.8 mm. ($1/32''$) for the 30 kgr./m. (60 lbs per yard) section, increases markedly with increased weight : it rises to 4.8 mm. ($6/32''$) excessive height for the 55 kgr./m. (110 lbs per yard).

The difference in question shows that the components of the sections cannot be calculated independently of each other; when two are given the third follows; for example, when p' and a' have been calculated from expressions 7 and 5, the lineal expression 6 will be $c' = h - (p' + a') = 0.41\ B + 23.65$, instead of the equation No. 6 : $c' = 0.608\ B + 19.76$. The problem to be resolved is to find the optimum proportions for the heights p', a', c' for a pre-determined height h. A similar consideration also arises in regard to the optimum distribution of the mass of the rail between the three parts, base, web and head; we shall revert to this in Chapter II.

It must be mentioned, in concluding this Chapter, that sections 1, 7 and 21 are English, 2, 3, 5, 9, 11, 12, 13, 15, 16, 18, 20, 24 and 27 French, 12 and 18 bull-head, 8 Bone-Guelma, 4, 10 and 22 American, 6 and 17 Swiss, 14 and 19 German, 23 Italian, 25 and 26 Belgian : they are all contained in the drawings appended to « Cours de chemin de Fer » by Descubes and Martinet.

3. *Conclusion.*

Whilst admitting that each rail section, from the original choice between bull-head and flat-bottomed, has a history justified by economic or technical factors, it must be conceded, from the preceding remarks, that it is necessary to achieve some uniformity, at the least within each country, between the various components of the sections; in this way, rails would be neither thin not thick unnecessarily, and never deformed; in brief, sections would be well-balanced.

Chapter II.

Details of 7 standard rail sections of the French railways, and 8 rail sections of the Greek State Railways, arranged in order of weight.

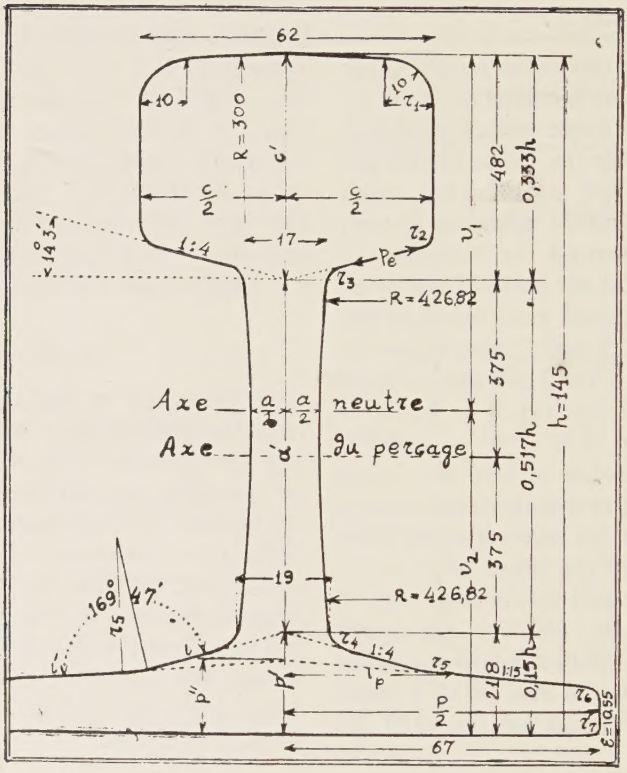
A. **Details of 7 standard rail sections of the French railways, from 20.86 to 55 kgr./m.**

1. *Drawings 3 and 4 of standard sections S_5 and S_6 of 46 and 50.586 kgr./m. (92.69 to 102 lbs per yard).*

These two drawings are reproduced *a*) to apply to No. 3 the symbols used in this

DRAWING No. 3.

Standard French flat-bottomed rail 46.02 kgr./m.



article and *b*) because they are reasonably well proportioned as regards the relationship between the components and the mass.

2. *Tables 5 and 5'.*—In these tables are, in increasing weight order, the weights corresponding to the 7 sections $S_1 \ldots S_7$ horizontally, and the components relating to each section in the columns $S_1 \ldots S_7$, each line will thus give the elements of the same heading p, p', a, a', c, c', ... etc. corresponding to each section.

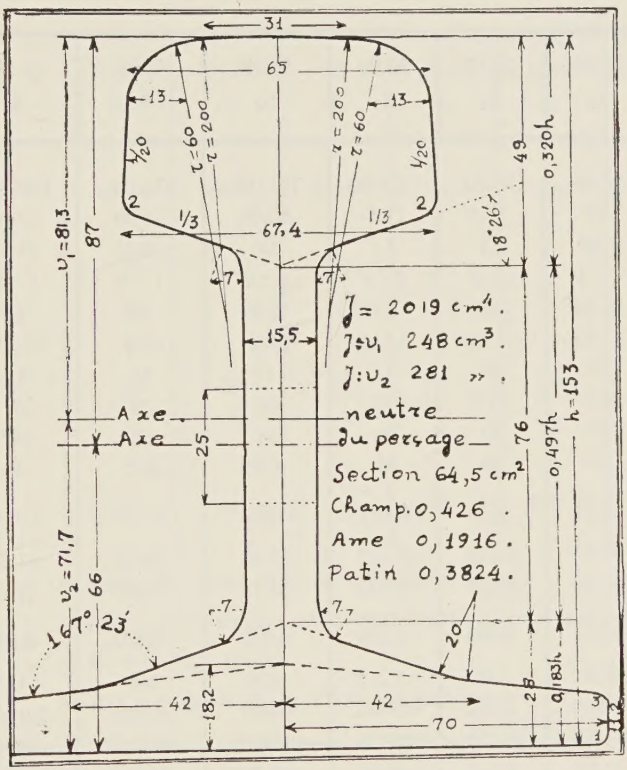
Table 5 includes the values which it has seemed advantageous to calculate in order to note the variation between one section and another, for example, the ratios $c' : c$, $v_2 : v_1$, $p : h$, $c' : h$, $a' : h$, $p' : h$, $a : a'$, $w_1 : w_2$, $s\,\text{cm}^2 = 0.2756\,h^2 \ldots$ $I = 0.1376\,h^2 s$, $I_1 = 0.2715\,hs$, $I_2 = 0.279\,hs \ldots$

3. *Graph No. 6.* — This graph represents the zig-zag lines derived from Table 5.

4. *Graph 6a*, represents the average lines proposed to replace the zig-zag lines of Table 6; Table 6_b and $6'_b$ indicate the

DRAWING No. 4.

New 50.586 kgr./m. S. N. C. F., rail for main lines.



analytical formulae for the lines of Table 6*a*. In Tables 5 ans 5′ have been noted, on each horizontal line, *maximum* or *minimum* coefficient or value, being marked with a *v* or a *dot*; it can thus be seen at a glance what the maximum and minimum limits are. Each average or zig-zag line is indicated by the corresponding symbol of drawing 3. Each average line passes through the points at the side of the symbol and outside the table at left or right. On the zig-zag the dimensions have not been inserted; the scales used for the ordinates are different, on the one hand to avoid confusing several zig-zag lines and on the other hand because it is the relative position only of the peaks which is of interest and not the value of any ordinate individually.

In the graph of average lines are noted the values of the ordinates for the extreme abscissae (20.86 and 55.18 kgr./m. [42 to 110.4 lbs per yard] for Graph 6*a*, 98.5 and 155 mm. [3.86 and 6.10 in.], for Graphs 7*a* and 8*a*).

TABLE No. 5

Characteristics of standard French rails of 20.86 to 55 kgr./m.

Nos.		20.86 S_1	26.10 S_2	30.00 S_3	36.09 S_4	46.02 S_5	50.586 S_6	55.185 S_7
1	p	90.00·	100.00	106.00	115.00	134.00	140.00v	134.00
2	p'	15·	17.3	17.5	20.05	21.80	28v	26.8
3	p''	7.50·	8	8.1	10	10.55	11.2	15v
4	i	1 : 4·	1 : 4	1 : 4	1 : 4	1 : 4	1 : 3v	1 : 4
5	i_p	21.34	33.6	32	34.59	38	42v	41.02
6	p_e	15.62	11.2·	15.65	14.2	14.2	20.78v	14.2
7	a	9·	10	11	13	15	15.5	19v
8	a'	54.5·	58.7	73	68	75	76v	75
9	c	50·	50	56	58	62	65v	62
10	c'	29·	34	35	39.95	48.2	49	53.2v
11	$\frac{c'}{c}$	0.58·	0.68	0.625	0.6896	0.777	0.754	0.858v
12	v_1	49.92·	56.45	61.957	65.65	74.20	81.3	82.8v
13	v_2	48.58·	53.55	65.543	62.35	70.80	71.7	72.2v
14	$\frac{v_2}{v_1}$	0.972	0.948	1.256v	0.943	0.954	0.882	0.872·
15	h	98.5·	110	125.5	128	145	153	155v
16	p/h	0.914	0.909	0.846.	0.899	0.924	0.915v	0.864
17	I	357.02·	551.51	836.96	1009.07	1620.10	2019	2186.50v
18	I_1	71.52·	97.62	136.09	153.70	218.34	248	264.07v
19	I_2	73.49·	102.9	131.72	161.84	228.83	281	302.84v
20	w_1	3.429·	3.74	4.536	4.259	4.746	4.903v	4.785
21	w_2	3.523	3.942	4.391	4.484	4.974	5.555	6.034v
22	s cm^2	26.74·	33.46	38.45	46.27	59	64.5	70.75v
23	s	$0.2756h^2$	$0.2765h^2$	$0·.2441h^2$	$0.2824h^2$	$0.2806h^2$	$0.2755h^2$	$0.2945h^{2v}$
24	δ	7800·	7800·	7802	7800·	7800·	7840v	7800·
25	I	$0.1376h^2s$	$0.1362h^2s$	$0.1382h^2s^v$	$0.1331h^2s$	$0.1306h^2s$	$0.1337h^2s$	$0·.129h^2s$
26	I_1	$0.2715hs$	$0.2652hs$	$0.282hs^v$	$0.2595hs$	$0.2552hs$	$0.2514hs$	$0·.241hs$
27	I_2	$0.279hs$	$0.2796hs$	$0.279\ hs$	$0.2733hs$	$0.2675hs$	$0.2849hs^v$	$0.2761hs$
28	i'	1 : 10v	1 : 15	1 : 15	1 : 15	1 : 15	1 : 10	1 : 15
29	ϵ	7.5·	8	8.1	10	10.55	11.2	15v
		20.86	26.10	30.00	36.09	46.02	50.586	55.185

TABLE No. 5'.

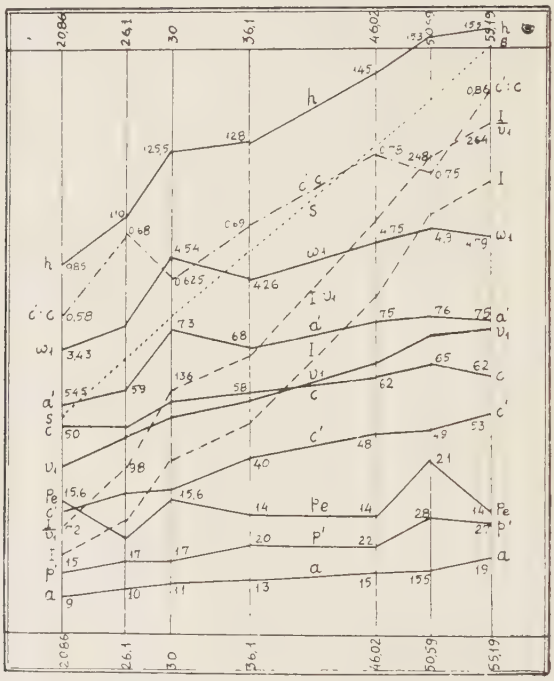
Standard French flat-bottomed rails.

	Weight →	20.86 S_1	26.10 S_2	30 S_3	36.09 S_4	46.02 S_5	50.58 S_6	55.185 S_7
		mm.	mm.	mm.	mm.	mm.	mm.	mm.
1	r_1	8·	9	9	9	10	13	8
2	r_2	2	2	2	2	2	2	2
3	r_3	6·	6.6	7.3v	6.21	7.21	7	6.19
4	r_4	6	5.6	7.8v	5.22·	6.21	7	6.19
5	r_5	28v	28	28	28	28	20·	28
6	r_6	3v	2·	2.7	2	3	3	3
7	r_7	0	1	1	1	1	1	1
8		180	260	0	270	300	200	300
9		r_1 and r_2 for tread and upper fishplate seating.						
10		r_3 and r_4 for web and upper and lower fishplate seatings.						
11		r_5 for two base slopes.						
12		r_6 and r_7 edge of base at top and bottom.						

5. *Tables 6_b and $6'_b$: Formulae for the average lines of Graphs 6 and 6_a as functions of B.* — The last column of this table indicates, for each increase or decrease of 1 kgr. per metre of weight B (+ or — 1 kgr./m.), the increase or decrease of the corresponding component of the section, or the quotient of the difference between the extreme values and the constant 34.34 (= 55.20 — 20.86); for example, for the height of the web a' of the rail, the extreme values are 30.87 and 50.22 mm. the difference 19.35 mm. related to the difference 34.34 kgr. of the corresponding weights 20.86 and 55.2 kgr./m., will give 19.35 : 34.34 = + or — 0.563 mm. increase or decrease of the height a' of the web of the rail; the inverse ratio + or — 1.775 kgr. for + or — 1 mm. of a' is worth noting; for the thickness a of the web it is found that for + or — 1 kgr./m. of weight + or — 0.218 mm. of the thickness of the web we should be allowed and, conversely, for + or — 1 mm. thickness of the web, we should have 3.44 kgr. weight per metre length of rail, or per km. of rail ≈ 8 tons of rail more; this is a serious consi-

GRAPH No. 6.

from Table 5 — Standard French 20.86 kgr./m. to 55.185 kgr./m. rails.



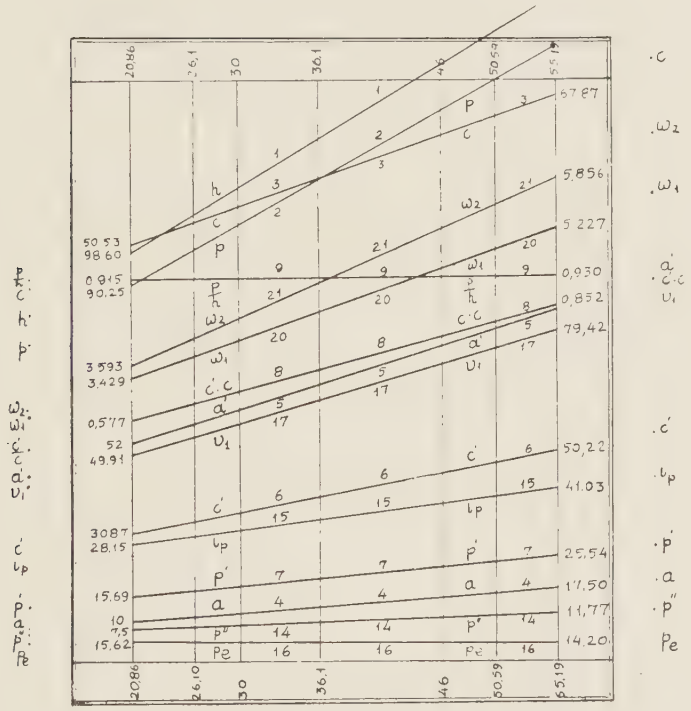
deration for economising in the thickness *a* of the web, without however going below 10 mm. nor exceeding 16 mm. except for rails in tunnels, where 19 mm. may be reached (standard section S_7). It must not be forgotten, moreover, that an increase of 1 mm. in the thickness *a* of the web leads to an increase of 2 mm. in the width *c* of the head so as not to reduce the rail clip bearing surface p_e, which caused a fresh increase in weight per metre.

6. *Table 7 and Graph 7a.* — This table and the corresponding graph have been arranged to square up the columns and those of the similarly-headed Tables 1 and 1′.

For certain of these components there is agreement between the two tables, for others there are divergences; moreover, the formulae for French standard rails have been drawn up starting at 20.86 kgr./m. and not from 30 kgr./m. as in Tables 1 and 1′; they will be different if compiled from 30 kgr./m.; the statistical or empirical formulae are to be applied only between the limits on which they have been compiled.

GRAPH No. 6*a*

average lines of Graph 6.



7. *Table 8 and Graph 8a.* — Graph 8*a* has been compiled by taking as abscissae the heights *h* of the sections S_1 .... S_7 and as ordinates the corresponding components of each section. It will be seen that the zig-zag lines of the components noted are less irregular than those of Graphs 2 and 2′ in Chapter I and of Graph 6 in this Chapter : consequently, the proposed average lines and their analytical expressions are preferable to those of Graph 6*a* and Tables 6*b* and 6′*b*.

In Table 8 have been included also the average formulae of Table 7, which are also expressed as functions of height *h*, including their average values as functions of ½ (maximum + minimum).

8. *Dynamic formulae of the 7 standard French sections.*

Taking into account the zig-zag lines for these sections, the following formulae may be used for a quick calculation of a reasonable approximation:—

1. *Sections S_1 to S_4* :

a) Moment of inertia :

$$I\ cm^3 = 42.81\ B - 526 \quad . \ . \ . \quad (28)$$

TABLE No. 6*b*.

Formulae for governing lines of Graphs 6 and 6*a* as functions of B.

Nos.	Formulae	Extreme values	For ± 1 kgr.
1	$h = 1.802B + 62$	98.6 — 160.5	± 1.802
2	$p = 1.690B + 55$	90.25 — 148.3	± 1.69
3	$c = 0.505B + 40$	50.53 — 67.87	± 0.505
4	$a = 0.24B + 5$	10 — 17.5	± 0.218
5	$a' = 0.951B + 32.2$	52 — 84.7	± 0.952
6	$c' = 0.564B + 19.1$	30.87 — 50.22	± 0.563
7	$p' = 0.287B + 9.7$	15.69 — 25.54	± 0.287
8	$\frac{c'}{c} = 0.008B + 0.41$	0.577 — 0.852	± 0.008
9	$\frac{p}{h} = 0.00075B + 0.889$	0.915 — 0.930	± 0.0407
10	$\frac{c'}{h} = 0.001427B + 0.264$	0.294 — 0.343	± 0.001426
11	$\frac{a'}{h} = 0.616 - 0.002346B$	0.567 — 0.487	∓ 0.00233
12	$\frac{a}{a'} = 0.001312B + 0.138$	0.204 — 0.210	± 0.000175
13	$\frac{p'}{h} = 0.154 - 0.0008B$	0.1524 — 0.1496	∓ 0.00008
14	$p'' = 0.124B + 4.9$	7.5 — 11.75	± 0.1243

TABLE No. $6'_b$.

Continuation of Table No. 6_b.

Nos.	Formulae	Extreme values	For $\pm$ 1 kgr.
15	$i_p = 0.375B + 20.33$	28.15 — 41.03	$\pm$ 0.375
16	$p_c = 16.48 - 0.0414B$	15.62 — 14.20	$\mp$ 0.04135
17	$v_1 = 0.5222h - 1.52$	49.91 — 79.42	$\pm$ 0.522*
18	$v_2 = 0.4778h + 1.52$	48.58 — 75.48	$\pm$ 0.476*
19	$\frac{v_2}{v_1} = 1.01 - 0.000387h$	0.972 ÷ 0.950	$\mp$ 0.00039*
20	$w_1 = 0.0523B + 2.34$	3.429 ÷ 5.227	$\pm$ 0.052
21	$w_2 = 0.0659B + 2.218$	3.593 ÷ 5.856	$\pm$ 0.0569

Formulae of governing lines of Graphs 6b and corresponding Table 5.

22	$p' : h = 0.159$ (5)	25	$p : h = 0.913$ (8)
23	$a' : h = 0.528$ (6)	26	$a : a' = 0.191$ (9)
24	$c' : h = 0.313$ (7)	27	$c' : c = 0.709$ (10)

* The values relate to $\pm$ 1 mm.

The numbers in parentheses are those of Table 5'.

TABLE No. 7.

Parallelism of rails in Tables 1 + 1′ and Table 5.

	B	h	c'	a'	p'	$c' : h$	$a' : h$	$p' : h$
1	20.86	98.5	29·	54.5·	15·	0.294	0.554	0.152
2	26.1	110	34	58.7	17.3	0.309	0.534	0.157
3	30	125.5	35	73	17.5	0.279·	0.581v	0.140·
4	36.09	128	39.95	68	20.05	0.312	0.531	0.157
5	46.02	145	48.2	75	21.8	0.333	0.517	0.150
6	50.59	153	49	76v	28v	0.320	0.497	0.183v
7	55.19	155	53.2v	75	26.8	0.343v	0.484·	0.173

	c	$c' : c$	a	$a : a'$	p	$p : h$	s	$\frac{I}{v_1}$
1	50	0.580·	9·	0.165	90·	0.914	26.74	71.52
2	50	0.680	10	0.170	100	0.909	33.46	97.62
3	56	0.625	11	0.151·	106	0.846·	38.45	136.09
4	58	0.689	13	0.191	115	0.899	46.27	153.70
5	62	0.777	15	0.200	134	0.924v	59	218.34
6	65	0.754	15.5	0.204	140v	0.915	64.50	248
7	62	0.858v	19v	0.253v	134	0.864	70.75	264.07

Average formulae as functions of B kgr./m.

$h = 1.802B + 62$ (1) $\qquad a' = 0.951B + 32.2$ (3)

$p' = 0.287B + 9.7$ (2) $\qquad c' = 0.564B + 19.1$ (4)

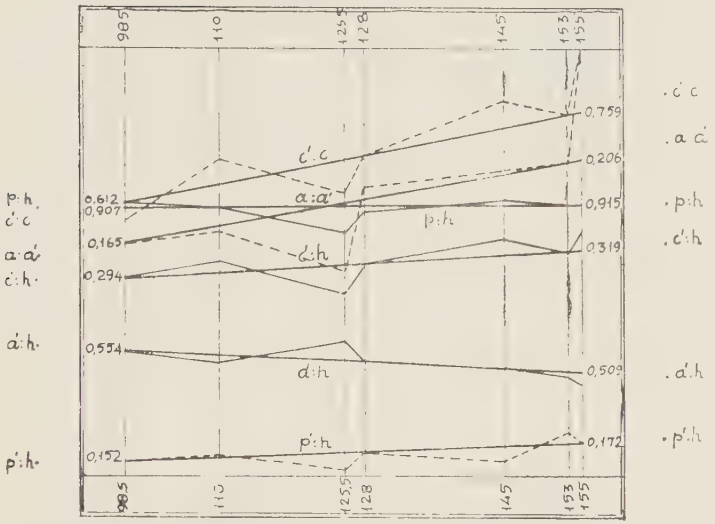
$p' : h = 0.159$ (5), $\qquad a' : h = 0.528$ (6), $\qquad c' : h = 0.313$ (7)

$p : h = 0.913$ (8), $\qquad a : a' = 0.191$ (9), $\qquad c' : c = 0.709$ (10)

h, p, p', a, a', c, c' in mm.

GRAPH No. 7*a*

relating to Table 7.



b) Moment of resistance :

$$\frac{I}{v_1}\ \text{cm}^3 = 5.396\ B - 4.1 \quad . \ . \ . \ (29)$$

c) Moment of resistance :

$$\frac{I}{v_2}\ \text{cm}^3 = 5.800\ B - 47 \quad . \ . \ . \ (30)$$

d) Coefficient of usefulness :

$$w_1 = 5.05\ B - 2.6 \quad . \ . \ . \ . \ . \ (31)$$

e) Coefficient of usefulness :

$$w_2 = 0.051\ B - 2.6 \quad . \ . \ . \ . \ . \ (32)$$

2. *Sections* S_1 — S_7 :

a) Moment of inertia :

$$I\ \text{cm}^4 = 62.95\ B - 1\,252 \quad . \ . \ . \ (33)$$

b) Moment of resistance :

$$\frac{I}{v_1}\ \text{cm}^3 = 6.505\ B - 81 \quad . \ . \ . \ (34)$$

c) Moment of resistance :

$$\frac{I}{v_1}\ \text{cm}^3 = 7\,382\ B - 105 \quad . \ . \ . \ (35)$$

d) Coefficient of usefulness :

$$w_1 = 0.0404\ B + 2.9 \quad . \ . \ . \ . \ (36)$$

e) Coefficient of usefulness :

$$w_2 = 0.054\ B + 2.5 \quad . \ . \ . \ . \ (37)$$

The formulae $1\ \text{cm}^2 = 0.1344\ h^2 s$ (38)

$$\frac{I}{v_1}\ \text{cm}^3 = 0.2641\ hs \quad . \ . \ . \ . \ . \ . \ . \ . \ (39)$$

and $$\frac{I}{v_2}\ \text{cm}^3 = 0.2749\ hs \quad . \ . \ . \ . \ . \ . \ (40)$$

can be used with a sufficient degree of approximation for sections S_1 and S_5. In these formulae h is expressed in cm. and s in cm^2.

TABLE No. 8.

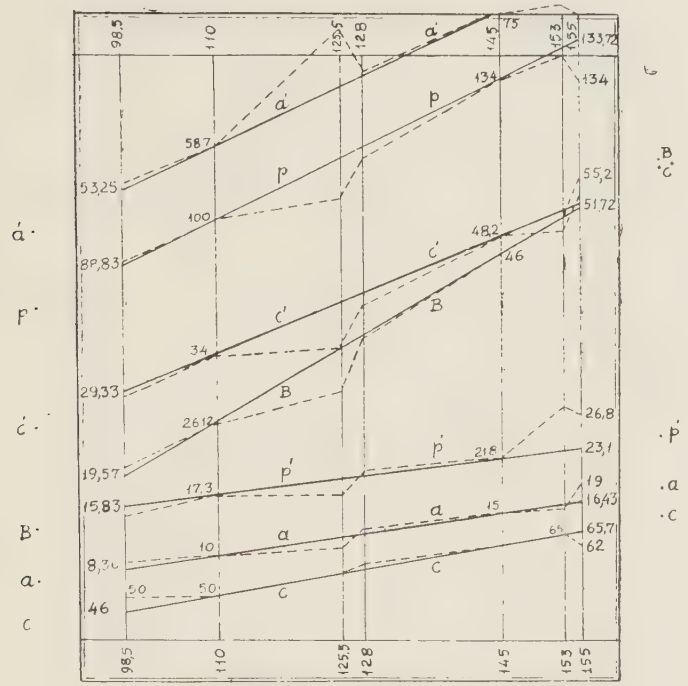
Formulae of main governing lines of Graph No. 8*a*.

Nos.	Formulae	Extreme values	For ± 1 mm. or ± 1 kgr.
1	$h = 1.758B + 64.10$	100.77 — 161.14	± 0.569
2	$p = 0.9714h - 6.85$	88.83 — 143.72	± 0.625
3	$c = 0.3488h + 11.64$	46 — 65.70	± 1.742
4	$a = 0.143h - 5.73$	8.36 — 16.43	± 4.252
5	$a' = 0.4657h + 7.48$	53.25 — 79.66	± 1.300
6	$c' = 0.406h - 10.66$	29.33 — 52.27	± 1.496
7	$p' = 0.1286h + 3.16$	15.83 — 23.09	± 4.725
8	$B = 0.569h - 36.471$	19.57 — 51.72	± 1.877 ½(Max.+Min.)
9	$p' : h = 0.159$	0.140 — 0.183	0.1615
10	$a' : h = 0.528$	0.484 — 0.581	0.5325
11	$c' : h = 0.313$	0.279 — 0.343	0.3110
12	$p : h = 0.913$	0.846 — 0.924	0.8850
13	$a : a' = 0.191$	0.151 — 0.253	0.202
14	$c' : c = 0.709$	0.580 — 0.858	0.7190

Arithmetical averages

GRAPH No. 8*a*.

relating to Table 8 — seven standard French sections.



9. *Table/Graph 9.* — This *table/graph* has been compiled, and is valid only for the range of the 7 sections S_1 ... S_7, so as to provide formulae, the graphs of which have no angular points. These formulae are reproduced below to give them a numerical order :

41. $s\ \text{cm}^2 = 1.2755\text{B}$ . . . . . (41)

42. $s\ \text{cm}^2 = 0.28h^2$ . . . . . (42)

43. $h\ \text{cm} = 2.137\sqrt{\text{B}}$ . . . . . (43)

44. $\text{B kgr./m.} = 0.219h^2$ . . . . . (44)

45. $\text{I cm}^4 = 0.0375h^4 = 0.78189\text{B}^2$. (45)

46. $\frac{\text{I}}{v_1}\ \text{cm}^3 = 0.0719h^3 = 0.70266\text{B}\sqrt{\text{B}}$ (46)

47. $\frac{\text{I}}{v_2}\ \text{cm}^3 = 0.0779h^3 = 0.76122\text{B}\sqrt{\text{B}}$ (47)

By following a different method of approach it is found that :

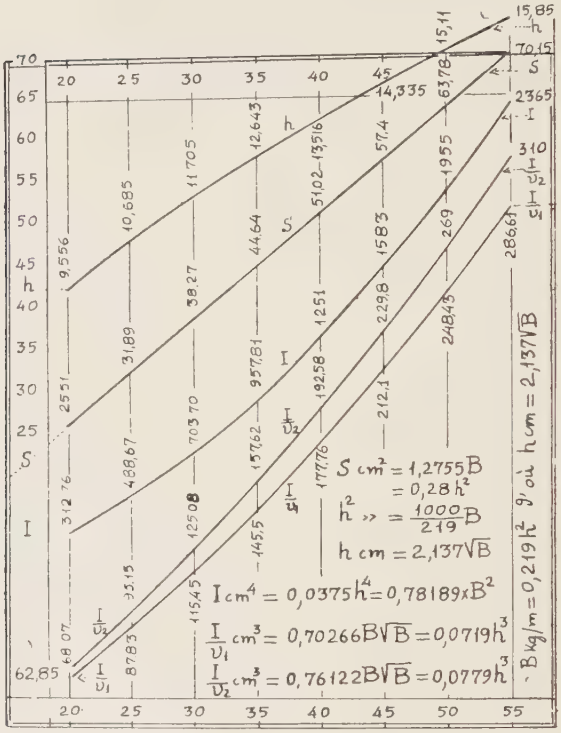
45′. $\text{I cm}^4 = 0.03733\ h^3$ . . . . . (45′)

46′. $\frac{\text{I}}{v_1}\ \text{cm}^3 = 0.07228h^3$ . . . . . (46′)

47′. $\frac{\text{I}}{v_2}\ \text{cm}^3 = 0.07517h^3$ . . . . . (47′)

GRAPH No. 9.

Continuous curves for rail section calculation.



A_1. Small Standard French fittings.

A close examination of the small standard fittings of the sections S_1 ... S_7 shows that they are difficult to determine for an intermediate section without definitely deciding on the outline.

The following particulars only may be noted :

1. The clip for section S_1 is 65.2 mm. high, 15 mm. thick and weighs approximately 3.25 kgr.

2. The clip for section S_2 is 72.5 mm. high, 20 mm. thick and weighs 5.8 kgr.

3. The clip for section S_4 is 83.02 mm. high, 26 mm. thick and weighs 8.83 kgr.

4. The clip for section S_5 is 91.7 mm. high, 30 mm. thick and weighs 11.21 kgr.

This clip is also used for section S_7.

The *bolts*, 4 in number, weigh :

0.287, 0.486, 0.510, 0.650 each for the 20.86, 26.30 and 36 kgr./m. sections, and 0.892 kgr. for the 46 and 55 kgr./m. sections.

The *coachscrews* weigh, each :

0.45 kgr. for the 20, 26 and 30 kgr./m., and 0.55 kgr. for the 36, 46 and 55 kgr./m. sections.

B. Details concerning 8 rails sections of the Greek State Railways, arranged in order of weight from 30 to 44 kgr./m.

1. *Table 10 and Graph 10$_a$.* — Table 10 and Graph 10$_a$ include the 8 sections in use on the various lines of the Greek State Railway system. Of these sections, Nos. 1 to 4 have long been used in Macedonia : the others are relatively new, the Belgian section No. 8 dates from 1926.

It is considered useless to stress these, at the same time, however, noting that they are, with the exception of No. 3, characterised by a high, massive base and the height a' of the web is rather small; the *plus* or *minus* variations of each section are shown clearly in Graph 10$_a$.

The resultant analytical equations, taking the heights h as abscissae, are as follows :

1. $B = 0.50h - 3$ or $h = 2B + 60$

2. $p = h - 28$ or $h = p + 28$

3. $c' = 0.2857h + 5$ 3'. $c = 0.322h$

4. $a' = 0.4857h + 2$ 4'. $a' = 0.503h$

5. $p' = 0.2286h - 7$ 5'. $p' = 0.175h$

6. $c = 1/3h + 17.334$

7. $a = 0.6667h + 3.67$

Conclusions regarding cases A and B.

The broken lines of the 7 standard French sections and the 8 Greek sections are less pronounced than those considered in Chapter I. *Revision of them would be advantageous.*

C. Considerations governing the constitution of a rail section.

In the foregoing, we have given details relating to existing rail sections, so as to show the limits between which they vary, and also to show their divergence from a statistical mean established within the various statements and graphs. We will now examine the *optimum profiles* obtained in each case, commencing with the base p of the rail.

1. *Base of rail. — Width p.* — The greater the width p of the base, the less will be the fatigue of the sleeper and the greater will be the moment of resistance to overturning. The bases of English rails are wider than the Continental ones, since they are mounted on softwood sleepers (Baltic pine). By the use of metal sole-plates, necessarily wider than the rail base, the sleepers are protected and the moment of resistance to overturning increased, but this represents a fairly considerable additional expense and results in a less resilient running, a more rapid wear of the base at the bearing surface on the sole-plate which precludes strengthening the track by reducing the average distance between sleepers, difficulty in fixing the rail to the sole-plate and sleeper, etc.

Moment of overturning $F \times h$. — Supposing a force F applied horizontally to the top of the rail, a moment of overturning of the rail $F \times h$ is given, which should be less than the moment of resistance to overturning $P \times \frac{p}{2} + A.p$, P being the weight per wheel on rail, and A the force required to pull out the coach-screws, ($\approx$ 3 800 kgr. per coachscrew),

TABLE No. 10.

Greek State Railways rail sections.

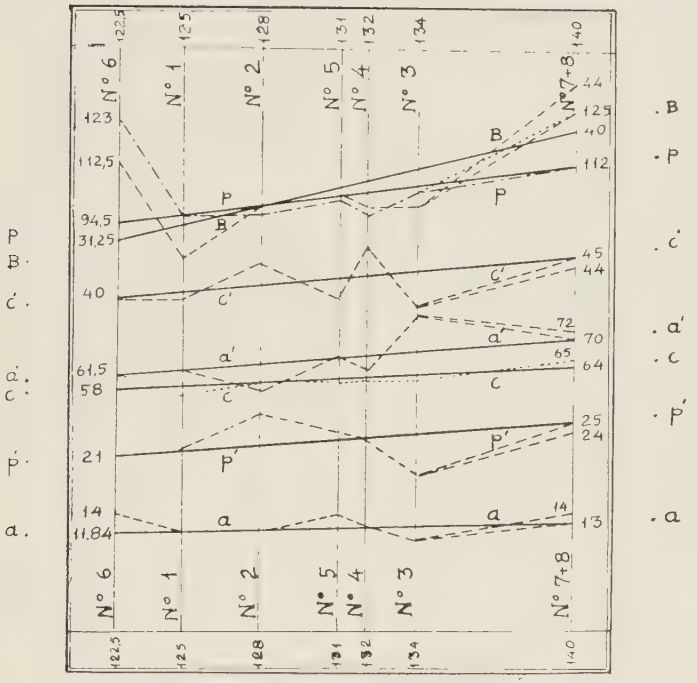
Nos.	B	h	c'	a'	p'	$c' : h$	$a' : h$	$p' : h$
1	30	125	40	63	22	0.320	0.504	0.176
2	34	128	44	58	26	0.344	0.453	0.203
3	34	134	39	76	19	0.291	0.567	0.142
4	34	132	46	63	23	0.349	0.477	0.174
5	35	131	40	66	24	0.313	0.506	0.181
6	37.5	122.5	40	61.5	21	0.326	0.502	0.172
7	40	140	44	72	24	0.314	0.514	0.172
8	44	140	45	70	25	0.321	0.500	0.179
Averages :						0.322	0.503	0.175

Nos.	c	$c' : c$	a	$a : a'$	p	$p : h$	s	$I : v_1$
1	57	0.702	12	0.190	100	0.800	38.46	127.57
2	60	0.733	12	0.210	100	0.781	43.59	137.00
3	60	0.650	11	0.144	105	0.783	»	
4	60.5	0.760	12.5	0.200	100	0.758	»	128.75
5	60	0.667	13.9	0.210	104	0.794	44.87	
6	61.8	0.647	14	0.227	123	1....	48.08	
7	64	0.687	13	0.180	112	0.800	51.28	194.3
8	65	0.692	14	0.200	125	0.893	56.41	192
Averages :		0.692		0.195		0.825		

Nos. 1 and 4 : ΘM and E.Θ.K. section.
» 2 : π. Δ. ε. section.
» 3 and 8 : Belgian section 1914 and 1926.
» 5 : A.ε. or Krupp section.
» 6 : A.ε Indian section.
» 7 : Carnegie section.

GRAPH No. 10*a*.

Greek State Railways rail sections.



without reckoning the resistance to overturning provided by neighbouring sleepers.

One should thus have :

$$F \times h \leqslant P \times \frac{p}{2} + A.p$$

$$\text{or } p : h \geqslant F : (0.5P + A)$$

that is to say, p must be as large as possible, all other things being equal; its values, as well as those of the ratio $p : h$ are contained in Tables 1′ (cols. 5 and 6) and 5 (1st and 16th lines), in Table 10 and its Graph 10*a* for the Greek State Railways. This ratio should be greater than 0.90 and approach as nearly as possible to 1.00, the value allowed by the Anglo-Americans (see sections 1, 7, 9, 10, 21 and 22 and Greek profile 6).

The wide bases are difficult to roll at a low temperature, they cool off more quickly than the remainder of the section, the homogeneity of the texture of the metal is impaired as the thicknesses ε, p' and p'' are insufficient.

The formulae for calculating the elements p, p′, p″ *and* ε *are* :

$$p = 0.914h \text{ ou } 1.7B + 54 \quad . \quad . \quad . \quad (1)$$

$$p' = 0.18h \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

$$p'' = \frac{2}{3} p' = 0.12h \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

$$\varepsilon = 0.073h \geqslant 10 \text{ mm.} \quad . \quad . \quad . \quad . \quad (4)$$

For radiusing corners, we propose $r_7 = 2$ mm., $r_6 = 3$ mm., and $r_5 = 30$ mm. The radius $r_7 = 1$ mm. is not adequate for manipulating the rails and avoiding splintering of the lower edges.

The 1 : 4 slope for the lower clip bearing surface and 1 : 9.72 $\approx$ 1:10 for the extension to the end of the base will be the most suitable.

The co-ordinates $x_0 y_0$ of the meeting points of the two slopes of the base i and i_1 are given by the equations

$$\frac{x}{\alpha} + \frac{y}{\alpha_1} = 1 \text{ and } \frac{x}{\alpha_1} + \frac{y}{\alpha_1 i_1} = 1,$$

i and i_1 being the two slopes, i of the clip bearing surface and i_1 of the slope to the edge of the base; it will be found that $x_0 = \frac{\alpha i - \alpha_1 i_1}{i - i_1}$ $y_0 = \frac{i i_1}{i - i_1}(\alpha_1 - \alpha)$; applying these to the section S_6 it is found that

$$a = 3 \times 28 = 84,\ i = \frac{1}{3},\ \alpha_1 = 182,$$

$$i_1 = \frac{1}{10},\ x_0 = 42,\ y_0 = 14.$$

2. *Web of rail. — Thickness* a. — This component is the one not the subject of calculation. However, the web fulfils two essential requirements, 1) it provides a solid connection, without bending or breaking, between the two main parts of the rail, i. e. base and head; 2) it must ensure the moments of resistance I : v_1 and I : v_2 in order that the fatigue of the metal of the head and of the base should not exceed a pre-determined value. In practice, it is necessary not to go *below 12-13 mm.* for the following 3 reasons :

1. too thin a web is difficult to roll.

2. too thin a web is not strong and could give rise to cracks, particularly the bolt-holes.

3. too thin a web cools off much more rapidly than the two elements it unites, which could give rise to a dangerous lack of homogeneity in the metal.

The following formulae : $a = 0.10\ h = 0.0306 \approx .\ 0.20\ a' \geqslant 12$ mm. . . (5) are those of section S_6, with $h = 153$ and B = 50.586 kgr./m. (see this section in drawing No. 4); they are the most suitable for avoiding the defect of a too thin web. It should be noted that an increase or decrease of the thickness a entrains a large increase or decrease in the weight per metre of the rail, as shown in Tables 2_a, 6_b and 8.

Height a' *of the web.* — The higher this is, the greater will be the moments of resistance. This principle was departed from when formerly very high rails were rolled, and these have nowadays been discarded.

The formulae :

$a' = 0.497\ h = 1.503$ B. . . . . (6), which are those of section S_6, may be applied.

Connections of web to head and base. — The connections of the webs of sections S_2, S_4 and S_5 are cylindrical with radii of 251, 357 and 427 mm. This practice is designed to increase the thickness of the web at its base and summit, whilst retaining the thickness a at the centre, given by the above formula No. 6. From the point of view of homogeneity of the metal at the top of the base and at the underside of the head, this practice is good, but it

has the drawback of reducing the clip bearing surface by the amount of additional web thickness and as the radii r_3 and r_4 are increased so the radii of the cylindrical surfaces are reduced : the cylindrical radius should not fall *below 400 mm.* nor should the radii of the joints r_3 and r_4 exceed 6 to 7 mm.

3. *Head of rail. — Width* c *of the head.* — This is the least variable component between different sections of normal gauge flat-bottom rails. In effect, normal gauge is defined as 1 500 mm. and rarely as 1 510 mm., between the rail centres taken squarely across the track; moreover, a minimum slackening of 10 mm. with new tyres and 25 mm. with tyres worn to a minimum is allowed; on the other hand, international traffic is allowed a distance of 1 425 mm. between the outer edges of wheel tyres, this gives

$$1\,425 \text{ mm.} + 10 + 2 \times \frac{c}{2} \leqslant 1\,500 \text{ mm.},$$

which shows $c \leqslant 65$ mm. or 75 mm. in the case of track 1 510 mm. between rail centres.

Section S_6 has a head width $c = 65$ mm. so as to increase as much as possible the upper clip bearing surface, sections S_5 and S_7 of 46 and 55 kgr./m. have a width $c = 62$ mm. only.

It is proposed to use the 65 mm. width which gives the large clip bearing surface, this width also being allowed in the Greek section No. 8 (Belgian, 1926 manufacture, B = 43.85 kgr./m.), it gives rise to less fatigue and less wear of rail and tyre treads.

It must not be overlooked that, as soon as the above dimension of 1 425 mm. and maximum slackening of 25 mm. are specified for international traffic, a new rail head with a width of 50 mm. would give, from the time it is placed in service, the maximum amount of play, i. e. 25 mm. In these circumstances, it could be that overworn tyres would bear on the rail heads almost at their edges (trailing vehicle tyres must cover at least 2/3 c, and motive units the full width c).

If, for reasons of economy, it is desired to use a smaller width c, and in relation to the weight B of the rail, formula $c = 1.285\ B = 0.425\ h$ —. . . . . (7) agrees with section S_6, it gives $c = 1.285 \times 50.586 = 0.425 \times 153 = 65.025$ mm.

It should be noted that the lateral faces of the heads of the Greek rails in Table 10 come noticeably close to the bottom of the web, (60 mm. width at the top and 50 mm. at the top of the clip), to provide a margin against lateral wear without regard to the more rapid wear of the clips and clip bearings as a result of the reduced area of contact. Almost all cases prefer *vertical* faces for the head; an exception is section S_6, which has lateral faces sloping towards the top so as to increase the clip bearing, to avoid rapid wear of the lateral faces, to economise in metal and to increase homogeneity by the adoption of two large junction radii, one of 60 mm. with the tread (300 mm. radius), the other 13 mm. to connect the second circle with the lateral face. The arrangement of the head in section S_5 thus satisfies fairly well the requirements of the more complicated head of section S_6.

Height c′ *of the head.* — This concerns a component which is fairly variable between one section and another. The columns $c' : h$, $c' : c$ of Tables 1 and 1′

and Graph 2′ prove this point. The ratio $c' : c$ of Table No. 5 as well as the other tables of the standard French sections and their corresponding graphs show fairly well the variations of this detail c' with that of h, c and B. We get :

$c' = \frac{1}{3} h$, $c' : c = 0.775$ for section S_5 and

$c' = \frac{1}{3.1} h$, $c' : c = 0.742$ for section S_6,

$c' = 0.332\ h$ with $c = \frac{1}{3} h + 17.53$ for the Greek sections.

The adoption of $c' = 0.325\ h$ or the ratio $c' : c = 0.760$ satisfies the conditions with the following reservation :

For heavily graded lines or lines with frequent braking, such as mountain lines, urban and suburban lines, lines adjoining garage roads and particularly large stations, the ratio $c' : c$ must be as large as is allowed by efficient rolling of the rails. In the same way, the head as well as the web and base of tunnel rails should be re-inforced; smoke, particularly sulphurous fumes, ashes, condensation of exhaust steam, inherent humidity inside long tunnels and poor ventilation attack rails more than braking does; the ratio $c' : c = 0.858$ for the 55 kgr./m. standard French rail (specially designed for tunnels) is obviously justified.

Tread of rail head. — It is generally acknowledged that this surface should be curved, not flat, so as to carry the weight transmitted by the wheels to the centre of the head and so avoid rapid wear of the head and also of the tyres. At the same time, there is no agreement on 1) the radius to be adopted for the tread and 2) on the radii of the function curves between the tread and lateral faces of the head. The tread curves for the standard French sections are shown on line 9 of Table 5. In the Greek sections, they are 200 mm. for sections 1, 2, 4, 5, 8 and 300 mm. for the 6th. It is logical to increase this radius with axle loads, or — what is really the same thing — with the weight B in kgr./m.; thus the formula R = 1.6 B + 215 corresponds to the standard French sections. It is also logical to reduce this radius with a reduction in the radius of track curvature, so as to reduce lateral wear of the faces in curves, as well as heavy tyre wear.

Radii of junction curves between tread and lateral walls of head. — Table No. 5 includes these radii, which range between 8 and 13 mm. It is considered that they should not be below 9 mm. nor above 10 mm., to avoid double contact, or — very much the same thing — the coincidence of the tyre and the upper edge of head. American steels permit the fixing of this radius at 8 mm., as is done in Greek section No. 6; the other junction curves of the Greek sections are — 12 mm. for sections 1, 2, 4, and 14 mm. for 5, 7 and 8. The use of a double junction radius, used also for section S_6 (see drawing No. 4) is recommended.

Chapter III.

German flat-bottomed rail.

Formulae and Graph No. 11 of 12 German rail sections from 4 to 50 kgr./m.

1. *Composition of formulae and Graph No. 11.* — On the basis of two tables

covering German flat-bottomed rails — the first giving the weight B in kgr./m., the corresponding height h in mm. and the moment of resistance $\frac{I}{v}1$ cm³ (see *Henschel Locomotive Manual*, 1935, p. 25) and the other giving, in addition to the foregoing components, the widths c and p of the head and base and the thickness a of the web, we are brought by the use of the zig-zag lines of the sections to the formulae shown below and represented by Graph 11.

The symbols used in these formulae are the same as those used in the preceding chapters. The scales used for the Graph are :

1. for the abscissae (weight) 2 mm. per kgr./weight;

2. for the ordinates, counted in mm. from the line 00 of the graph, full size (1 : 1) for the dimensions c, p and h; 5 mm./mm. (5 : 1) for the thickness a of the web; 0.5 mm. per unit of moment of resistance $\frac{1}{v_1}$ cm³; for the coefficient of usefulness w, 20 mm. per unit of w.

(N. B. — *The original drawing is reproduced about half size.*)

1. *Continuity formulae.*

a) *Base* p.

$p = 3.625\ B + 23.5$ . . . . (1) for B from 4 to 12 kgr./m.
$p = 2.000\ B + 43$ . . . . . (2) » » » 12 » 31 »
$p = 1.111\ B + 70.56$ . . . (3) » » » 31 » 50 »

Ratio $p : h$ — width p of base to height h of rail :

$p : h = 0.828$ for B 4 to 20 kgr./m.
$p : h = 0.816$ » » 20 » 50 »

There is a minimum $p : h = 0.692$ for B = 6 kgr./m.
and a maximum $p : h = 0.986$ for B = 12 kgr./m.

The average of this ratio is $p : h = 0.8234$ for rails of 4 — 41 kgr./m. The rail types of 39, 45.6 and 49.4 kgr./m. have the widest bases (= 125 mm.) in relation to their heights 136, 142 and 148 mm., which give values 0.919, 0.88 and 0.844 for the ratio $p : h$.

b) *Web.*

The web thicknesses a, in mm., correspond to the formulae :

$a = 0.344\ B + 2.625$ . . . (4) for B 4 to 20 kgr./m.
$a = 0.111\ B + 7.280$ . . . (5) » » 20 » 33.5 »
$a = 0.1935\ B + 4.52$ . . . (6) » » 33.5 to 50 »

c) *Head.*

The head width c in mm. corresponds to the formulae :

$c = 2.67\ B + 7.33$ . . . . (7) for B 4 to 10 kgr./m.

$c = 1.30\ B + 21$. . . . . (8) » » 10 » 30 »

$c = 0.368\ B + 49$ . . . . (9) » » 30 » 50 »

d) *Rail height.*

The rail height h in mm. corresponds to the formulae :

$h = 3.125\ B + 39.5$ . . . (10) for B 4 to 20 kgr./m.

$h = 2.300\ B + 56$ . . . . (11) » » 20 » 30 »

$h = 1.217\ B + 88.5$ . . . (12) » » 30 » 50 »

e) *Moment of resistance.*

The moments of resistance $\frac{I}{v_1}$ cm^3 correspond to :

$\frac{I}{v_1}$ cm. 3 $= 2.75\ B - 4.50$. (13) for B 4 to 10 kgr./m.

$\frac{I}{v_1}$ cm. 3 $= 4.20\ B - 19$. . (14) » » 10 » 20 »

$\frac{I}{v_1}$ cm. 3 $= 5.97\ B - 54.4$. (15) » » 20 » 50 »

f) *Co-efficients of usefulness.*

These co-efficients are, by definition, the quotients of the moments of resistance, given by formulae 13, 14 and 15, divided by the weights B :

$w = 2.75 - 4.5/B$ . . . . (16) for B 4 to 10 kgr./m.

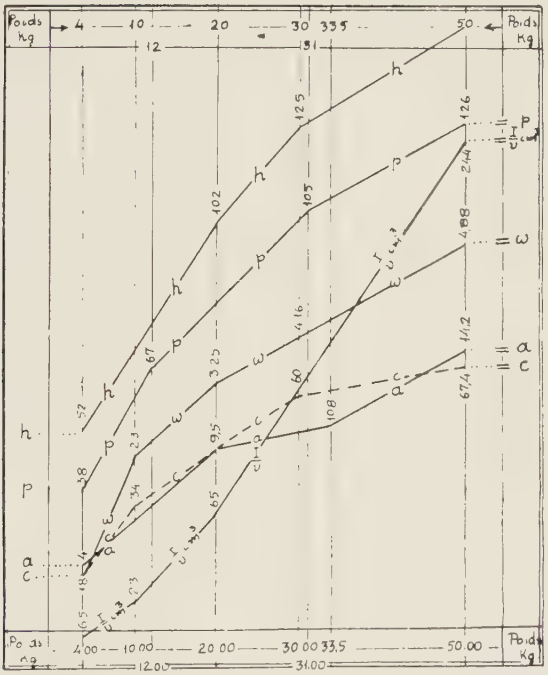
$w = 4.20 - 19/B$. . . . . (17) » » 10 » 20 »

$w = 5.97 - 54.4/B$. . . . (18) » » 20 » 50 »

or $w = 2.16 + 0.061B$ . . . (19) » » 20 » 50 »

GRAPH No. 11.

German flat-bottomed rails.



Poids = weight.

g) *Relation of s and B.*

Equal in section *s*, the German rails are heavier than the French rails because they have a specific weight $\delta = 7.86$ kgr., that of the French rails being $\delta = 7.84$ kgr., or 0.9974 of the former.

Consequently $s \times 0.786 = B.$. (22)

and s cm$^2 = 1.2722$ B. . . . . (23),

in which formulae s is expressed in cm^2 and B in kgr.

Note. — Of equal weight, the German rails increase in height at the expense of the base width *p* and the web thickness *a*, in order to provide a greater moment of resistance $\frac{I}{v_1}$ cm^3.

The practice of a wider base and thicker web is preferable in consequence of the greater resistance to overturning and the greater resistance in avoiding cracks in the web and breakages of the base.

Conclusion. — The rails in question give zig-zag lines more irregular than those of the rails previously considered. The sections grow in height to increase, for equal weight, the moment of resistance $\frac{I}{v}$, to

TABLE No. 12.

German normal gauge rail sections.

	B kgr.	h mm.	p mm.	c mm.	a	s mm²	$\frac{v_1}{v^2}$ %	I cm⁴	$\frac{I}{v}$ cm³	$p : h$ %	I : vB
1	33.4	134	105	58	11	4253	99	1036.6	154	78	4.61
2	33.8	130	104	58	13	4350	102	978	148.9	80	4.41
3	34.87	135	105	58	13	4471	97	1072	157	78	4.5
4	36.05	130	105	58	14	4600	101	1006	156	89	4.33
5	36.2	129	105	60	»	4620	107	1002	150.2	81	4.14
6	43	138	110	72	»	5230	97	1351.6	193.4	80	4.71
7	43.86	140	125	65	»	5587	106	1457	202	89	4.61
8	45.5	144	110	72	»	5739	98.6	1583	216.8	76	4.82
9	46.3	147	130	66	»	5907	101	1700	230	88	4.97
10	48.89	148	125	67	»	6228	106	1781	234	84	4.73
11	49.05	148	125	67	»	6248	103	1797	237	84	4.83
12	40.77	138.4	113.5	63.7	13.5	5203	101.6	1342.2	189	82.4	4.64

The nos. 1, 6, 8 : Prussia-Hessen-Schwerin-Oldenburg.

» 7 Bavaria-Württemberg-Baden, the no. 3 Bavaria.

» 2 Württemberg, the nos. 4, 9 Saxony.

» 5, Baden, the nos. 10, 11 Reichsbahn.

TABLE No. 13.

Continuity formulae for German normal gauge rails.

Nos.	Continuity formulae.	Minim.	Maxim.	Average.
1	$h = 1.18B + 90.116$	129	148	138.5
2	$p_1 = 0.833B + 75.84$	104	110	105,8
	$p_2 = 1.377B + 57.459$	110	125	120.8
3	$c = 0.59B + 38.06$	58	72	63.7
4	$a = 14$	11	14	13.54
5	$s = 127.476B - 4.698$	4253	6248	5203
6	$\frac{v_1}{v_2} = 0.00419B + 0.825$	0.971	1.03	1.016
7	$I \quad cm^4 = 53.705B - 837.23$	978	1797	1433
8	$\frac{I}{v} \quad cm^3 = 5.777B - 46.36$	148.9	237	189
9	$\frac{p}{h} = 0.00262B + 0.712$	0.76	0.89	0.82
10	$\frac{I}{vB} = 5.777 - (46.36 : B)$	4.14	4.97	4.61
11	$B = 33.4 \div 49.05$ kgr.			

Lengths h p_1 p_2 c and a in mm.

Areas s in mm^2.

Weights B in kgr./m.

Ratios $v_1 : v_2$ and $p : h$ in %.

the detriment of the ratio $p : h$, which suffers by default. They are given a specific weight of 7860 kgr./m³ instead of 7 840 kgr./m³ of the standard French section of 50.586 kgr./m.; the result is that for equal area, the German rails are heavier than the French.

3. *Rails of German Railways, of normal gauge, having weights of* 33.4 *to* 49.05 *kgr./m.*

It has been considered desirable to draw up Tables 12 and 13, relating specially to rails of the various German systems.

Table No. 12 is, with some modifications in classes, in the 26th edition of HÜTTE, volume 3, p. 895. Table 13 gives analytical continuity expressions which result from Table 12.

The Hütte Manual also gives, on p. 896, volume 3, rail sections of various countries from 43.2 kgr. (Sweden) to 65.3 kgr. (Pennsylvania); the following averages have been extracted :

B kgr.	h mm.	p mm.	c mm.	a mm.	I cm⁴	I : v cm³	$\frac{p}{h}$ %	$\frac{I : v}{B}$
56	151	133	68	15.3	2005	256.5	88	5.0

These correspond very nearly to the standard French section of 55.7 kgr./m.

Chapter IV.

Formulae used for calculating rail resistances.

The problem to be resolved can be set in several different forms; one can determine, for example, first the maximum load P kgr. of the wheel on the rail, which is taken as on fixed supports at a distance of l cm., the maximum fatigue is also evaluated, to which a rail should be subjected in the static state and it is desired to find the moment of resistance $\frac{I}{v}$ cm³ against bending moments to which the rail is subject on the passage over it of wheels. This can, however, also be based on a pre-determined moment of resistance and on a maximum wheel load and the average distance l cm. of consecutive supports fixed, so as not to exceed a maximum fatigue decided in advance. In the first case, the unknown quantity to be found is the moment of resistance $\frac{I}{v}$ cm³; in the second case it is the average distance l cm. It is the second problem which most often arises.

1. *Expert opinions of Profs. Sevène and Vicaire.*

In his E. P. C. set course, the well-known Prof. SEVÈNE wrote, that the most accurate rail calculator was past experience. The other expert, VICAIRE, Professor of the Ecole Nationale Supérieure in Paris, supported Prof. SEVÈNE, opinion, in another way; to calculate for each rail the quantity $l \times \frac{v}{I}$ for various values of

l and on rails in service which were approved by the Permanent Way Dept. and to see how, by comparison, modifications should be made in l or $\frac{v}{I}$ or both together, to satisfy a new weight P per wheel. Prof. VANDERRYDT and M. MINSART (Engineer) in their course « Railway Operating », Prof. MARTINET in his course for the « Ecole Nationale des Ponts et Chaussées », the Belgian engineer, M. DESPRETS, in his learned work on the question, the research in the United States, all confirm in our opinion the opinions of SEVÈNE and VICAIRE, in contributing to evolution by experience. If each formula gives results satisfactory to the user, this is due to the high factor of safety allowed for the steel of the rails, which may be 8-12 kgr. mm² (5.079-7.619 tons per sq. inch.) for a metal which has a resistance to fracture by bending of 60, 70 and even 80 kgr./mm² (38.09, 44.44 and even 50.79 tons per sq. inch.). The actual fatigue R of the metal is related to many factors, both of track and of stock, which are so variable that it is not possible to submit them to full calculation.

2. *Formulae for rail calculations most generally used.*

a) French formulae : There are two of these :

$$R \times \frac{I}{v} = 0.199\ Pl \approx 0.20Pl \quad . \quad . \quad (1')$$

$$\text{»} \quad = 0.25Pl_2 \quad . \quad . \quad . \quad . \quad . \quad (2)$$

$$R \times \frac{I}{v} = \frac{4}{27}Pl_1 = 0.148Pl_1 \quad . \quad . \quad (1')$$

$$\text{»} \quad = \frac{1}{4}Pl_2 = 0.25P_2 \quad . \quad . \quad . \quad . \quad (2')$$

Formulae Nos. 1 and 2. — In compiling formula No. 1, consideration has been given to a rail section between the axes of two consecutive supports as a beam of span l, (without connection with the remainder of the rail) resting freely on these supports. This hypothesis, the most unfavourable for the rail, gives a maximum bending moment $M_1 = \frac{1}{4}Pl_1 = 0.25\ Pl_1$ in the section at the centre of the truss in question and is produced on the passage of a wheel.

A more unfavourable hypothesis would be set up if we suppose that the rail section considered is bedded in the two supports (still without any connection with the remainder of the rail); in this event the most heavily stressed sections will be those held at the support, and this will be produced when the loads are a distance of $1/3\ l_1$ from the two supports. The fixing moments are then equal, in absolute value to $M_1 = \frac{4}{27}Pl_1 \approx 0.148\ Pl_1$. Providing for an average case between the two hypotheses, we get a bending moment

$$M_{moy} = \frac{1}{2}\left[\frac{1}{4} + \frac{4}{27}\right]Pl_1 = 0.199Pl_1 = 0.2Pl_1$$

As regards the overhang at each end of the rail, this has been assumed to be disconnected from the remainder of the rail and fixed on its bearing point; the maximum moment would be on the fixed section when the load $\frac{P}{2}$ is located at the end of the portion, the other half being carried by the opposite piece, assumed fixed on its support. The bending mo-

ment would then be $M_2 = \frac{P}{2}\frac{l_2}{2} = 0.25$ Pl_2, l being the distance between axis of the bearings of the joint. By bringing the joint sleepers nearer together, the bending moment M_2 is decreased until it is equal to M_{moy} (*).

These are the hypothetical bases for the application of formulae 1 and 2.

Formulae 1' and 2'. — No explanation of the hypothetical establishment need be given in view of the explanations given of the hypotheses for the preceding cases 1 and 2 : we simply exclude the foregoing unfavourable case of the freely supported truss, taking only the case of full fixing, which comes back to the acceptance of a maximum bending moment of $M_{max} = 0.148\ Pl$, instead of $M_{moy} = 0.2\ Pl_1$. The moment M_{max} is a little lower than 3/4 of the the moment M_{moy} ($M_{max} : M_{moy} = 0.74$).

For the parts overhanging the joints, the two cases are based on the same hypothesis, i. e. fixing on the joint sleepers and load $\frac{P}{2}$ at the end of the rail sections instead of P; the distance l_1 of the joint sleepers will be $\approx 0.60\ l_1$ in Formula 1'.

For the application of these formulae, the French railways accept a fatigue R of 850 kgr./cm² (12 089.82 lbs per sq. inch.) for static loads and an increase of 50%, or 1 275 kgr./cm² (18 134 lbs per sq. inch.) for live loads.

There is no doubt that the simple fitting on sleepers hypothesis would be realised only for an instant and in the case where the distance of the centres of the two consecutive wheels is equal to l_1, which would not occur even with goods engines with a minimum driving wheel diameter ($\approx$ 1 metre), provided that the length l_1 does not exceed 0.90 m. (2'11½"). For more potent reasons, fitting in will be impossible because of the wheelbase of normal vehicles, which is much larger than l_1.

Moreover, the free support hypothesis will be more false as the width of the bearing increases; its minimum value is 120 mm. (4 23/32") for normal gauge track.

b) *German formulae.* — Two formulae are used :

$$R\frac{I}{v} = 0.187 Pl \approx 0.19 Pl. \quad . \quad . \quad (3)$$

and $$R\frac{I}{v} = 0.25 Pl \frac{7 + 8\gamma}{10 + 4\gamma}. \quad . \quad . \quad (4)$$

known as the ZIMMERMANN formula.

Formula No. 3. — Winkler's formula.

It is assumed, for the establishment of this formula, that the rail portion previously considered is bedded on one support and set free on the other; by means of this hypothesis the maximum bending moment $M_{max} = 0.1924\ Pl$ will occur when the load P is 0.423 l distant from the bearings, it will be $M_{mim} = 0.187\ Pl$ when the load is applied to the middle of the truss; the average case is allowed :

$$\frac{1}{2}(M_{max} + M_{min}) = 0.1897\ Pl \approx 0.19\ Pl.$$

which formula differs very little from French formula No. 1.

For the application of this formula a stress of 1 000 kgr./cm² is allowed.

(*) $M_{moy} = M_{average}$.

Formula No. 4.
Zimmermann's formula.

This formula has been established by assuming a length of rail, separated from the line and resting freely on 4 consecutive supports placed at the same level. On support No. 2, the bending moment

$$M_2 = \frac{1}{4} Pl \frac{3 - 4\gamma}{10 + 4\gamma} \text{ or } \gamma = \frac{6EI}{csl^3}, \text{ E}$$

being the elasticity of the steel, I the moment of inertia of the section of the rail, c the ballast coefficient, i. e. the number of kgr. necessary, so that when laying 1 cm² (0.155 sq. inch.) of ballast, a depression of 1 cm. is produced, s the bearing surface of a half sleeper, and l the average distance between sleepers. The bending moment in the middle of the span 2 — 3 when a load P is applied, will have an absolute value of :

$$M_{23} = \frac{Pl}{4} \times \frac{7 - 8\gamma}{10 + 4\gamma} = \frac{Pl}{4} \mu.$$

The definitive formula is:

$$R \times \frac{1}{v} = \frac{Pl}{4} \mu \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

where μ is an increasing function of γ and consequently, to reduce the bending moment M_{23} it is necessary to reduce γ, which can be done by increasing the ballast co-efficient c and secondly by reducing the distance l between the supports, but without obtaining the full advantages from it, given that this element increases, by the reduction, the factor μ.

For further details of this formula, see the work of Vanderrydt and Minsart, p. 360, Vol. 1; Parodi and A. Tetrel, p. 268, and Zimmermann's work on this formula.

This formula provides a good guide for strengthening track, but the number of factors, which it is necessary to take into consideration, makes its use complicated.

For this reason the simple formulae 1, 2, 1′ and the Winkler No. 3 are the most used.

c) *Calculation of weight P in the United States.* — In the U. S. A. formulae for rail calculation also allow a moment of resistance $\frac{I}{v}$ cm³ as well as a weight B kgr./m. [see, for example, the formula for French standard profiles :

$$\frac{I}{v} \text{ cm}^3 = 0.70266 \sqrt{B.} \quad . \quad . \quad . \quad (46)]$$

To this purpose the following steps are taken :

Divide into classes : 1. *light rails* of less than 30 kgr./m.; 2. *medium rails* of 30 — 45 kgr./m.; 3. *heavy rails*, over 45 kgr./m. Moreover, the sleeper spacing allowed is $l_1 = 0.726$ m. and $l_2 = 0.45$ m. for joint sleepers. Subject to this it is allowed that a rail of B kgr./m. can sustain an axle load P, equal to :

P = 457.2 B for light rails,

P = 548.64 B for medium rails, or 20% more than (1),

P = 640.08 B for heavy rails, or 40% more than (1).

For rails of weight B : 30, 45 and 50 kgr./m., we find :

$P_{30} = 13.72$ t.; $P_{45} = 26.29$ t.; and $P_{50} = 32$ t. per axle, with an average sleeper spacing of 72.6 cm. (2′4 39/64″).

With american rails of 66 and 76 kgr./m. we find : $P_1 = 42.245$ tons and $P_2 = 48.646$ tons per axle.

3. *Application of the more usual formulae for the calculation of the weights P/wheel or rail heights* h.

a) *Table No. 14.* — This table has been drawn up to demonstrate that it is not necessary to be restrictive as regards the formulae used. Thus the Administrations which use the French formula No. 1′ of complete insetting, find a weight P_2 35.13% higher than the weight given by formula 1 of the average bending moment between simple bearing and insetting; certainly, the difference is not small.

Similarly, in Germany, despite the choice of a coefficient 0.19 of the product $P_3 l$, instead of the 0.20 product of $P_1 l$, the German formula No. 3, following the adoption of a steel fatigue of 1 000 kgr./cm² (14 223 lbs per sq. inch.) instead of 850 kgr./cm² (12 089 lbs per sq. inch.) as allowed by the French railways, gives a weight of 23.84% more. It gives only 0.9164 of the weight given by the French formula, that is $P_3 = 0.9164\ P_2$, which rather justifies the practice of those systems using French formula No. 1.

According to these results, it is necessary to allow in the formula $R \times \frac{I}{v} = KPl$ for resistance, the three factors, $\frac{I}{v}$, P and l representing the pre-determined figures, whereas the other two factors — rail fatigue R and the moment which produces it — KPl are unknown as a result of the uncertainty of the value of the numerical co-efficient K. Prof. VICAIRE was on good ground when he proposed to establish a statement of the values he ratio $R : K = Pl \times \frac{v}{I}$ to ascertain the limits between which the ratio varies with rails which have been satisfactory in use, with dimensions P, l and $\frac{v}{I}$. In Table 14, we have multiplied the U. S. A. practice dimensions by the relation $\frac{72.6}{l}$, to recall that it is always necessary to take this into account each time it differs from unity : thus, 1 46 kgr./m/rail would have a weight $P = 648.08 \times 46 = 29\,811.68$ kgr., but for a sleeper spacing $l = 72.6$ cm.; if this distance were 90 cm. we should only have $29\,812 \times \frac{72.6}{90} = 24\,058$ kgr.

b) *Table No.* 15. — This has been compiled to meet the question : « given particulars P (kgr./wheel), l average sleeper spacing in cm., R permitted rail stress per cm², what is a suitable rail height h? »

The same table gives the two formulae :

$$h \text{ cm} = 0.1482\sqrt[3]{Pl} \quad . \ . \ . \quad (a)$$

$$h \text{ cm.} = 0.1381\sqrt[3]{Pl} \quad . \ . \ . \quad (b)$$

(*a*) for French rails and (*b*) for German rails. The second column of the table shows the sleeper spacing l cm. : in the other columns the headings show loads P in *tons per wheel*, the numbers of the columns give the heights corresponding to P and l; for example, a wheel load of 10 tons and a sleeper spacing of 75 cm. (2′ 5½″) has a suitable rail height of 134.7 mm. for French rails. This height corresponds to a weight B kgr./m = $0.219\ h^2 \simeq 40$ kgr./m.

From the eighth line of the table can

TABLE No. 14.

Application of more usual formulae for flat-bottomed rail.

Formulae used		Section	Distance between sleeper axes in cm. 90	80	70	60
			Weight per axle in tons.			
French formulae	N° 1 $850 \frac{I}{v} = 0.2\ P_1 l$	S_1	6.75	7.60	8.68	10.13
		S_2	9.22	10.37	11.85	13.83
		S_3	12.85	14.46	16.53	19.28
		S_4	14.52	16.33	18.66	21.77
		S_5	20.62	23.20	26.51	30.93
		S_6	23.42	26.35	30.11	35.13
	N° 1 $850 \times \frac{I}{v} = 0.148\ P_2 l$	S_7	24.94	28.06	32.06	37.41
		We have $P_2 = 1.3513\ P_1$				

German formula :

$1000 \frac{I}{v} = 0.19\ P_3 l$ — We have $P_3 = 1.2384\ P_1$

U. S. A. formulae

$P = 457.2B \times \frac{72.6}{l}$ for light rails

» $= 548.64B$ » for medium rails

» $= 640.08B$ » for heavy rails

1st formula up to section S_3 (30 kgr./m.)

2nd formula up to section S_5 (46 kgr./m.)

3nd formla beyond section S_5

TABLE No. 15.

Height h of rail, in cm., as a function of load Pt/wheel and l cm between sleepers.

Nos.		Height of rail for P tons =						
		5	6	7	8	9	10	11
1	90	11.36	12.07	12.70	13.28	13.81	14.31	14.77
2	85	11.14	11.84	12.47	13.03	13.55	14.04	14.49
3	80	10.92	11.60	12.22	12.77	13.28	13.76	14.20
4	75	10.69	11.36	11.96	12.50	13.00	13.47	13.90
5	70	10.44	11.10	11.69	12.22	12.70	13.16	13.58
6	65	10.19	10.83	11.40	11.92	12.39	12.84	13.25
7	60	9.92	10.54	11.10	11.60	12.07	12.50	12.90
8	5	2.40	2.55	2.68	2.80	2.91	3.02	3.11

From the general formula $R \cdot \frac{I}{v} = KPl...$ assuming successively :

$R = 850$ kgr./cm^2 $K = 0.2$ et $R = 1000$ kgr./cm^2 $K = 0.19$ and $\frac{I}{v}$ cm^3 $= 0.07228\ h^3$.

For the two cases :

$$h \text{ cm} = 0.1482 \sqrt[3]{Pl...} \text{ (a) and } h \text{ cm.} = 0.1381 \sqrt[3]{Pl...} \text{ (b)}$$

where P is the load per wheel in kgr. and l is the sleeper average spacing in cm.

The heights h cm. have been calculated by the formula (a) the 0.931 of these heights from formula (b).

From the figures on line 8 it will be seen that to each decrease of 5 cm. of l corresponds a decrease of h mm., going from 2.4 to 3.11 mm.

easily be calculated the *economy* or *expense* of metal which results from the decrease or increase of l and these results can be compared with the increased or decreased expense involved in the purchase of sleepers.

General conclusion.

In studying the statistical data of this article, it is essential to recognise :

That the various components of the section of a rail, linear, circular and surface dimensions are not always well proportioned, thus :

1. The rail weight B in kgr./m. is excessive or deficient in relation to its height, in the case of thick or thin rails.

2. The weight B kgr./m. is not always effectively distributed between the base, web and head of the rail; this is to increase the moment of resistance.

3. Similarly, the height h of the section is sometimes badly proportioned as regards base, web and head (sometimes a very high web).

4. The radii of junctions are sometimes excessive (head) or deficient (base) without good reason.

For these four reasons, we propose :

As regards 1) :

$$h \text{ cm.} = 2.137 \sqrt{B} \quad . \quad . \quad . \quad (43)$$

or B kgr./m. $= 0.219h^2$,

contained in Table/Graph 9 and in the text under (43) and (44).

As regards 2) : 32% for the base, 20% for the web, 48% for the head.

In the U. S. A. the area of the section is arranged as 38% base, 22% web and 50% head.

From Prof. Vanderrydt's work we get :

36.8 + or — 3.7% base.

18.6 + or — 3.7% web.

45.5 + or — 4.5% head.

As regards 3) : the proportions 18%, 50% and 32% of h for heights p' of the base, a' of the web, and c' of the head, are those of section S_6 (50.586 kgr./m.) which are recommended.

As regards Greek profile No. 7 (Carnegie), this has the following distribution : 34.05% base, 19.55% web, 46.4% head.

As regards 4) : a radius of 300 and of 200 mm. (11 13/16″ and 7 7/8″) for the tread and for large and small radius lines of the outline [the first for $R \geqslant 500$ m. (1 640′ 5″) and the second for radii less than 500 m.] is indicated. The junction radii r_1 between the tread and the side walls of the head will be between 8 and 10 mm., 8 mm. $\leqslant r \leqslant$ 10 mm. (5/16″ and 25/64″) not forgetting that the radius connecting the tyre tread with the flange is about 15 mm. (19/32″). A double radius of 20 (25/32″) and 10 mm. would be preferable, as is used in S. N. C. F. section S_6 and in the Greek sections. For radii r_5 and r_7 the values of 30 (1 3/16″) and of 2 mm. (5/64″) are justified, in preference to those of the French (20 and 1 mm.) (25/32″ and 3/64″). Finally between the slopes 1 : 3 and 1 : 4 for the clip bearing, the second is preferable : it increases the bearing surface for the fishplate and does not stress the bolts so highly.

***

Note. — It is recalled that the data and conclusions of this Article refer to rail sections of a maximum weight of 55 kgr./m. which agrees with systems, other than those of the U. S. A., where the maximum axle load is 22 tons.

In the U. S. A. axle loads have long exceeded 28 tons. The standard rail of 66 kgr./m. (133 lbs per yard), and also that of 76 kgr./m. (153 lbs per yard) of the Pennsylvania Railroad are capable of supporting, the first 42 and the second 48.65 tons per axle. In these conditions, the dimensions and masses, both absolute and relative, naturally vary from those in this Article. The following figures refer to them : $c = 76$ and 76 mm., $c' = 44$ and 47 mm., $c' = 0.2421h$ and $0.23h$, $c' : c = 0.58$ and 0.618, $h = 182$ and 204, $p = 153$ and 172, $a' = 0.577h$ and $0.66h$, $a = 17$ and 17 mm., $p' = 0.181$ and $0.11h$, $p : h = 0.84$ and 0.82.

The weight distribution of the 76 kgr./m. rail is about 28.05 kgr./base, 23.93 kgr./web and 24.02 kgr./head. These figures differ very considerably from those used outside the U. S. A. The first figures relate to 66 kgr./m. rail, the second to 76 kgr./m. rail.

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Wheel-rail contact.

Geometrical study — The envelope method,

by M. BOUTELOUP,

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(Revue Générale des Chemins de fer, October 1947.)

Engineers have always been deeply interested in the contact between wheel and rail, either from the standpoint of the strength of materials or from that of double contact. Under the former comes local fatigue of the elements in contact (described by Hertz in 1882) and under the latter comes double contact (more accurately second contact) which from geometrical considerations may take place either before or behind the point of support of the wheel. It results in friction, wear, sharp flanges and a tendency on the part of the wheel tyre to climb up, on to the rail with risk of derailment. (See reports by Péchot and Bernheim to the Technical Committee, *1900-1906.)*

The present note is concerned with the geometrical aspects of contact. It is the sequel to numerous researches undertaken in France and elsewhere and brings before investigators a simple and complete method for the determination of all possible elements of the second contact together with its position and angular relation to the rail.

The problem, and the method.

When a wheel pair is resting on the rails, if the axle is at right angles to the track all possible contacts lie in a vertical plane containing the centre line of the axle, and their study is a simple matter.

If, however, the position taken up by the axle is slightly oblique, there may be other contacts in front, or behind. What are they? Where are they? In order to provide an answer, we have recourse to a device. Let us imagine that we swing the whole assembly of wheel and rail through one complete revolution about the centre of the axle. The wheel will continue to coincide with itself, while the rail will sweep out a figure in space that will remain entirely external to the wheel since the rail does not at any point penetrate the wheel. In particular the surface of revolution forming the limits and the envelope of the space thus swept out will be completely external to the wheel and can have no points in common with it other than those of tangency; these latter will moreover appear on the two meridians.

The study of the contact of two solid bodies in space (tyre and rail) thus becomes the study of the contact of two profiles in one plane viz: the profile of the tyre and the meridian of the envelope.

It will therefore be of interest to determine this meridian.

The exact solution.

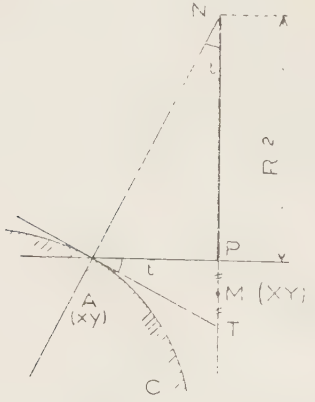
Investigation of the envelope is simplified by the fact that the curvature of the track is negligible in relation to that of the wheel, and that the rail is thus a cylindrical surface generated by straight lines pivoted in the transverse section of the rail. During the rotational movement, each generatrix pro-

duces a hyperboloid (a very flat hyperboloid, with one surface). The meridians of these hyperboloids are hyperbolas, and the envelope of the meridians is the meridian of the envelope. It is this which we are to determine.

A brief examination shows that this envelope closely resembles a hyperboloid with one surface. Only one part of this pseudo-hyperboloid is of interest to us, namely, the portion turned towards the flange of the wheel; and within this portion, another one smaller still — the throat, within the limit set by the height of the flange (36 mm. [1 $^{27}/_{64}$"]). It is this effective portion which we shall now examine in detail.

The determination of the envelope is effected by the elementary analysis, set out in *Appendix I.*

The calculations have an interesting peculiarity: they lend themselves to immediate geometrical interpretation and allow of a simple geometrical construction which gives the envelope point by point and the direction of the tangent at each point.



Fundamental construction.

Fig. 1.

The said construction, which is a fundamental of the present study, depends upon one sole parameter Ri^2, the product of the radius of the wheel and the square of the obliquity of the axle to the track (otherwise the tangent of the angle of shear of the wheel with respect to the rail). It is as follows (fig. 1):

At each point of the tyre the horizontal and normal are set up and a subnormal of height equal to Ri^2 is marked off. The point of the envelope is on this vertical. It lies midway between the horizontal and the tangent to the tyre.

The tangent to the envelope is parallel to the tangent to the tyre.

The radius of curvature of the envelope is equal to the sum of the radius of curvature of the tyre and the value of $\frac{Ri^2}{\cos^3 t}$, t being the inclination of the tangent to the tyre with respect to the horizontal.

The envelope is thus a simple anamorphosis of the profile of the rail.

It is moreover apparent from fig. 2 that the two points A and M of the rail and the derived curve respectively, are reciprocal; *the profile of the rail and its anamorphosis are two conjugate and reciprocal curves.*

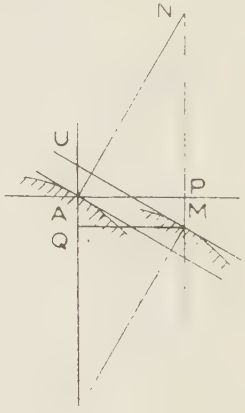


Fig. 2.

It will be specially noted that the *construction depends solely upon the argument* Ri^2, the length of which is equal to

the radius of the wheel multiplied by the square of the obliquity that it takes up: the value of this product *assumes special importance in each individual case.*

The meridian is the same for wheels of different radii (R) when the permissible obliquity varies in the inverse sense and gives the same value for Ri^2.

Notes. — The *geometrical solution* outlined above takes account only of the position of the rail with respect to the axle. *It is equally applicable to the guard rail* and its contact with the inner face of the tyre as it is to the running rail and the tread of the tyre. The only requisite is the meridian deduced from the profile of the guard rail.

The solution is also of immediate application to *tramway rails*, and when it is required to inscribe the profile of a wheel flange within the groove, the two meridians between which the flange must lie are derived from the two faces of the groove.

Plates 1 and 3 show a number of meridians set up for a wide range of values of the argument Ri^2. The first plate relates to the S.N.C.F. standard 46 and 50 kgrs. (102 and 110 lbs.) rails with a corner fillet of 13 mm. ($^{33}/_{64}$") radius. The running surfaces of these two profiles differ only in the large radii, and the differences are negligible. The second plate covers the rail with sunken flangeway IC2 (56 kgr. ([124 lbs.]) designed for curves, which is an international standard.

All these meridians bear a striking family resemblance to the well known profiles of worn tyres.

We will now examine more closely the choice of values for Ri^2.

Properties of the meridian.

(1) As previously stated, the *meridian* is necessarily *external to the profile of the tyre*, since all points on the rail are external to the tyre. The only common points are *points of tangency.*

(2) If there are points in common between the meridian and the profile of the tyre, it means therefore that there are *several points of contact* between the rail and the tyre. In practice there will be found meridians and tyres capable of contact at several points because the tyre is more concave than the meridian. There will also be others that can make contact at one point only because the respective profiles of rail and tyre are such that at all points of like orientation the radius of curvature of the tyre is greater than the radius of curvature of the meridian.

(3) A simple calculation (see *Appendix II*) shows that in the two conjugate and reciprocal curves, there is a simple relationship between the radii of curvature of corresponding points. The radius of the enveloping (or transformed) curve is equal to that of the enveloped curve (the rail profile) augmented by the value of $\frac{Ri^2}{\cos^3 t}$:

$$r_m = r_a + \frac{Ri^2}{\cos^3 t}$$

We have here the *means of designing a tyre* which, for a given radius of wheel and an assumed angle of incidence i, would never allow of double contact so long as the surfaces were new.

(4) *Limits of play in grooved rails.* — When a wheel pair is running between two rails at a given obliquity i, the limits of side play are determined by the displacement required to bring them away from contact with the right hand rail and into contact with the left hand one; or, more precisely, the distance between contact with the right hand meridian and contact with the left hand one. By passing the traced outline of the tyre over the relevant meridians or transformed rail curves for different values of Ri^2, *the reduction of sideplay imposed by the obliquity of the axle may be immediately ascertained.*

This fact is of direct application to the

accommodation of wheels in *grooved rails*, and permits measurement of the amount by which obliquity of the axle reduces:

— the play of a wheel from one side to the other of the same groove;

— the play of the wheel pair between the outside rails;

— the play when the wheels approach the two checkrails.

This is particularly interesting in regard to tramways where the flangeway is much narrower and the curves much sharper than in railway practice.

(5) *Would a « transformed curve » of the tyre profile be of interest?* — We have just seen that from a rail profile an enveloping or transformed curve may be derived which enables us to assess or to design a tyre profile, and that the two outlines are reciprocal. Is it possible, starting from a tyre profile, to

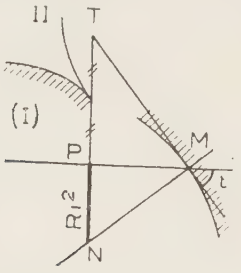


Fig. 3.

deduce a transformed « enveloped » curve and so obtain a rail profile of which the tyre itself is the transformed outline, and which might eliminate certain double contacts? In principle the construction is possible but in fact it is found that if the customary tyre profile is adopted and the design of the transformed curve is commenced from the top, that is to say from the running surface (fig. 3) a point is soon reached at which the transformed curve turns back upon itself (II) and a branch is formed which bears no relation whatever to the problem in hand. This results when the inclination t to the vertical becomes sharp enough to give:

$$\cos^3 t \leqslant \frac{Ri^2}{r_m} \quad . \quad . \quad . \quad . \quad . \quad (8)$$

where r_m is the radius of curvature of the tyre and t is the inclination of its tangent to the horizontal.

Such a result might well be foreseen since it has been shown that among the properties of the meridian (3rd property) we have:

$$r_m = r_a + \frac{Ri^2}{\cos^3 a} \quad . \quad . \quad . \quad . \quad (9)$$

whence:

$$r_a = r_m - \frac{Ri^2}{\cos^3 a} = r_m - r_m = 0$$

This means that the transformed tyre curve cannot serve as a rail profile and that its descending outline cannot be followed unless the tyre profile itself has at all points a radius of curvature

$$r_m > \frac{Ri^2}{\cos^3 t}.$$

A circular arc does not satisfy this condition. It is essential that the tyre outline should follow a pseudo-parabolic curve, which may moreover be a straight line as in the existing profile.

True position of the point of contact.

It is easy to define with precision the position in space of the point of contact between rail and tyre (see fig. 4). The point of contact which can be brought into coincidence with the point M of the meridian lies in reality on the generatrix AP of the rail. It is projected through P and is located in front of or behind the figure at a distance equal to $\frac{AP}{i}$, say 10, 20 or 50 AP if the

obliquity of the axle to the track is 1/10, 1/20 or 1/50. This distance is readily obtained by choosing a sub-normal P'N' = R*i* whence AP' = R*it*, which is the distance from the point of contact in space.

several arcs of bell curve becoming successively flatter, with angular points of transition (see Plate 1).

For other values of R*i* the abscissae of these curves all vary proportionally to R*i*.

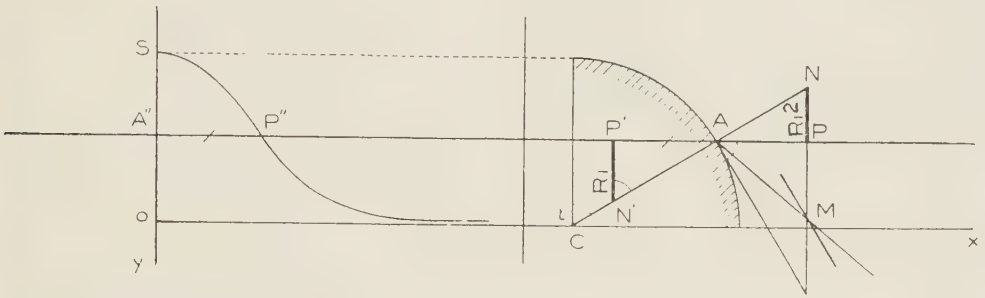


Fig. 4.

By setting out all the lengths AP' as A''P'' starting from one transverse profile S*y* of the rail, the *locus of possible contacts* may be constructed corresponding to one single value of R*i*.

If the head of the rail were formed of one cylindrical surface of revolution one would in this way obtain a bell-shaped curve asymptotic to the straight line *ox* which passes through the centre, and defined algebraically either by the system of equations:

$$y = -r(1 - \cos t)$$

where r is the radius of the head of the rail, and

$$x = \mathrm{R}i \operatorname{tg} t$$

R, i, being the radius and obliquity of the wheel; or, after eliminating t, by the cartesian equation

$$x = \frac{\mathrm{R}i\sqrt{-y\,(2r + y)}}{r + y}$$

Since the head of the rail is made up of several cylindrical arcs tangential to each other and having successively smaller radii, the locus of the contacts is formed from

The curves cannot be utilised throughout their length. The useful portion is limited to the point reached by the highest flange (36 mm.) expressed as a function of the radius of the wheel, thus (fig. 5):

$$x^2 = b\,(2\mathrm{R} + b)$$

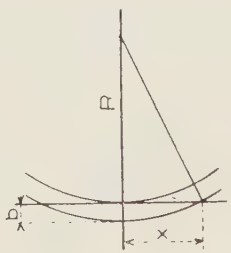


Fig. 5.

In the case of a flange b = 36 mm. and a wheel of diameter 2R equal to:

2R =
850 — 1 000 — 1 500 — 1 750 — 2 000 mm.

the length of contact:

x =
179 — 190 — 235 — 253 — 271 mm.

It will be noted that the meridians (Ri^2)

and the loci of contact (Ri) do not correspond each to each but each to all, and conversely. For a wheel of radius R at an angle of incidence i the meridian Ri^2 and the curve of contacts Ri must be chosen independently of each other, selecting a value appropriate to the requirements of the case.

Having determined the position of the point of contact, we are now in possession of *all the geometrical elements* essential to the study of the contact, namely:

— the point of tangency M of the two meridians;

— the generatrix A of the rail along which effective contact takes place, and its angle of inclination t;

— the distance A''P'' from the contact, or the arm of the frictional lever.

Numerical illustrations and scale of meridians.

Obliquity of wheels relative to track (angle of attack). — It has been shown that the whole trace of the meridians, or transformed rail curves, depends upon the argument or length of Ri^2 in question, R being the radius of the wheel and i the attainable inclination of the plane of the tyre to the rail, or angle of attack. What are the admissible values for this obliquity, in practice?

The obliquity of the wheel (angle of attack) may result from four factors :

— the play j of the wheel pair with respect to the track, variations in the distance between rails, (overspacing and wear included) and the distance between tyres (wear included) permitting an inclination

$$\frac{j}{L}$$

(where L is the greatest distance between flanged wheels (fig. 6).

— the play j' of the axleboxes in the horns, from front to rear of the vehicle, limited by the spring hangers and permitting an inclination $\frac{j'}{e'}$ where e' is the lateral spacing of

Fig. 6.

the axleboxes (Fig. 7). To this must be added occasional out-of-square errors in wheel mounting;

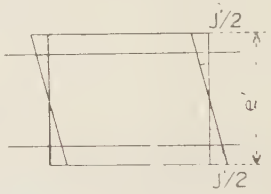


Fig. 7.

— end play of the journal in the bearing augmented by lateral play between the axlebox and its guide, permitting an inclination $\frac{j''}{L}$ where L is the wheelbase (Fig. 8);

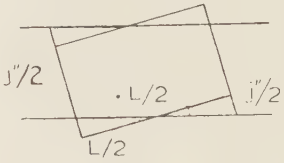


Fig. 8.

— curvature of the track, giving for a curve of radius C an obliquity $\frac{L}{2C}$ (Fig. 9).

There may in certain cases be greater inclination in locomotive straight axles placed outside the fixed wheelbase (0-5-0 engines of the South Eastern region).

To this angle of attack due to the curve

must be added in exceptional cases that due to kikns in the track and to the slight variations in the gauge that not even the most attentive supervision will prevent. Apart

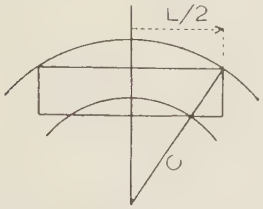


Fig. 9.

from the possibility of dangerous defect, the probability of this factor appears to be less than one in a thousand.

If all these elements of play become additive at any moment, notably while the vehicle is yawing, the obliquity of the axle may attain a value i :

$$i = \frac{j - j''}{L} + \frac{j'}{e'} + \frac{L}{2C}$$

It is seen that the length of the vehicle appears below the line in the first term and above it in the last. The case of subsidiary tracks, with their particular angle of deviation, would call for separate treatment.

The table in *Appendix IV* shows provisional figures, subject to verification, covering a series of characteristic cases for standard track (railways, the Metropolitan, and tramways) for metre gauge track (railways only), and for 60 cm. track of which only a brief outline is given.

We shall consider in each case the extreme limits within which the problem is encountered: the broadest gauge between wheels and the narrowest track permitted by the Berne regulations (or for metre gauge track and for grooved rails by the ministerial decree of July 8th 1908 which is rather unsatisfactory (*) and not universally honoured): also the narrowest gauge of wheels, badly worn, on the widest and most badly worn track.

In connection with curves, we have considered:

— for *standard track* :

a 250-*metre* radius (820'), common in mountain track, in Auvergne at all events,

a 150-*metre* radius (492'), frequently encountered in subsidiary track,

the 75-*metre* radius (246') (and sometimes even less) of the Metro,

the 50-*metre* radius (164'), occasionally met with in urban lines, cited here as an exaggerated instance, and

the 18-*metre* radii (59') (there are plenty even of 15-metre [49' 2 1/2"]) employed by standard-track tramways on routes used by vehicles having a wheelbase as great as one-fifth of this);

— for *metre gauge track* :

a 100-*metre* radius (328'), permitting speeds as high as 30 to 40 km./h. (18 to 25 m. p.h.),

a 50-*metre* radius (164'), and the exceptional one of 30-*metres* (98' 6") met with when crossing junctions.

Consideration might equally well be given to the 15-*metre* radius (49' 2 1/2") of the urban tramways;

— for 60 *cm.* (1' 11 5/8") *track* :

the 30-*metre* radius (98' 6") frequently seen in open routes permitting a speed of 12 m.p.h. and the 20-*metre* radius

(*) In a number of instances this decree stipulates bare dimensions only without tolerances for wear, which is impractical. In others, it gives one limit only. It would be in the interest of technical unity if this document were withdrawn.

(65' 7") employed on slow speed routes through villages.

Amongst *rolling stock*, we have chosen short bogies, capable of great angular displacement but having a moderate angle of attack on curves, and long vehicles, coaches, locomotives with large wheels, and railcars, which show less tendency to yaw but have a large angle of attack on curves.

So far as concerns the *play of the wheels* with regard to the frame, whether it be the longitudinal freedom allowed by the spring rigging and the horn cheeks, the lateral play of the axle journals in their bearings, or that between the axleboxes and the horns, we are much less well informed. We believe that the regulations governing workshop repairs vary considerably and that for wagons and coaches they are in some cases rough and ready.

The reference to 60 cm. track, which is rare in France, chiefly concerns military installations which we handled during the 1914-1918 war and which inspired the beginnings of the present study.

The Table therefore introduces suggestions rather than certainties. We shall welcome any criticism, any contribution of observed facts that our colleagues in French or foreign railways may care to offer, in order to fix trustworthy values for the angle of attack and thereby render the present study really objective.

An appropriate range of values of Ri^2 for the study of the envelope meridians, is 0.5 — 1 — 1.5 — 2 and 4.5 mm.

Constructional details of the chart.

The portion of the rail profile in which we are interested comprises a circular arc of large radius, varying from 100 mm. to infinity and forming the running surface, together with a corner radius the usual value of which is 13 mm. for standard track, 8-9 mm. for metre gauge, 6 or 5 or 3 mm. respectively for 60 cm. track, the grooved rail and its check rail. Theoretically, it is possible to envisage the absence of any radius in the case of a rail supposedly worn sharp. Between the crown of the profile and the corner radius there is usually an intermediate arc of from 25 to 60 mm. radius.

For practical purposes, the diagram should extend a little beyond the height of the flange of the wheel, say 30-36 mm. It is strongly recommended that a scale of 10/1 be chosen since this is particularly convenient and avoids calculation together with errors of interpretation. The diagram is thus 30 to 36 cm. in each dimension.

The running plane is marked off together with the normal to it and the tangent to the crown of the rail which is inclined 1 in 20 like the centre line of the rail and the tread of the tyre. The use of squared paper is a supplementary aid. The following construction lines should be allowed to remain: the radii C_1, C_2, C_3 of the corner radius of the rail head and the corresponding medians such as d_2 on which the points M'_2, M''_2, M'''_2 are aligned in series. *Construction* proceeds in fact *in series*, for when the point M''''_2 has been located (it corresponds on median d_2 to $Ri^2 = 2$), the points M'_2, M''_2, M'''_2 are immediately determined and are found to correspond to $Ri^2 = 0.5$ — 1.0 — 1.5 which appear at one-quarter, one-half and three-fourths of the length of the median.

Use of the chart.

A *transparent tracing of the tyre* under consideration, drawn to the same scale, is laid over the chart. It is marked out in squares to ensure accurate registration, and shows the running plane and the 1 in 20 tangent line, well produced.

It should also for convenience show the horizontal reference line, 10 mm. below the running circle, on which graduations showing the gauge between tyres are conventionally

marked. The reference line will be marked to show this spacing, the values of which are worthy of note:

1 410 — 1 426 — 1 435 — even 1 470.

These are in fact the limiting values for the narrowest gauge or the most badly worn wheels, the widest gauge of wheels, and the narrowest or the widest rail spacing. The first-named limit marks the degree of wear beyond which tyres have to be put back for re-turning.

A similar reference line is drawn on the rail profile.

The above-mentioned graduation shows clearly for track of given width how far the wheels may move transversely before they foul a meridian Ri^2. It will be seen that the obliquity of the wheels with respect to the track augments their sideplay by the quantity $\frac{1}{2} Ei^2$ (E is the rail spacing) which attains the following values :

where i =

30	40	50	60	70	80	90	100

with E = 1.00 $\frac{1}{2} Ei^2$ = (mm).

0.45	0.80	1.25	1.80	2.45	3.20	4.05	5.00

and with E = 1 435 $\frac{1}{2} Ei^2$ = (mm.)

0.64	1.14	1.80	2.58	3.52	4.60	5.81	7.17

On the other hand, the same obliquity will reduce the play by a similar amount with respect to the guard rails.

It is of interest to sketch in *an observed worn tyre profile* in order to ascertain which type of rail contact it represents and what loss of metal is involved in relation to the original profile. In the course of the methodical investigation proposed in the Conclusions, it would be possible to record the number of cycles of contact required to produce the successive stages of wear observable in any given wheel.

Grooved rails. — When dealing with grooved rails it is of interest at the same time to study the position of the second tyre in its groove. When the first tyre approaches the running rail the second one approaches the check rail. Either one may touch first, depending upon the relative spacings of the rails and the wheels. The design may be assessed by the aid of two cross-sections of the rail and the meridians Ri^2. These sections are set out with a chosen distance (*) between, and over them a transparent tracing of two flanges is laid, similarly spaced. One rail section and one tyre tracing will suffice if the latter is first placed in its mean position, bearing in mind the assumed dimensions of the track and tyre gauge, and this position is marked for reference. The tyre is now brought towards the rail, noting its position with respect to the meridians. It is then displaced in the opposite direction by the same amount, towards the check rail, being meanwhile regarded as the mirror image of the second tyre in the second groove. The values of Ri^2 available to the tyres without their bearing upon either the rail or the check rail are then immediately apparent.

Examination of an objection to the envelope method : lifting of the wheel. — The following question has been put : Is the envelope method valid if the wheel should lift? Evidently so. If the wheel lifts with or without making contact with the rail, the axle assuming an obliquity i(**), we may always imagine the rail being revolved about the centreline of the axle. It will sweep out a figure in space which will remain completely external to the wheel and with which the wheel will either make no contact or only tangential contacts.

This contact may be studied by bringing together the tyre profile, supposedly raised, and the envelope meridian constructed from

(*) The spacing will be equal to or slightly less than $V-2.Or$, V being the gauge of the track and Or the width of the groove.

(**) Which ,by virtue of the lift, may increase slightly.

suitable values of y or of R, that is to say values augmented by the lift of the wheel. The length of yi^2 or Ri^2 will be slightly increased thereby, which implies *in effect an advance by a known amount* in the predetermined scale of meridians.

Remarks about the reciprocal wear of rail and tyre.

I. — Each encounter between rail and tyre is equivalent to the stroke of a file which wears both members. A wheel of radius R meeting a rail at different angles of incidence i receives as many strokes (in so far as there is relative slip) as if it were making effective tangential contact with a file having the corresponding profile of Ri^2.

The tyre must necessarily wear much faster than the rail. For instance, if it has a circumference of 3 metres, each point on the tyre will make contact more than 300 times per kilometre and more than 30 000 times in a daily journey of 100 kilometres. The tyre of a coach in the long-distance train running 1 115 kilometres daily between Paris and Ventimiglia will make 350 000 contacts.

A rail on a busy route over which 30 trains of 100 wheel pairs each may pass in 24 hours, will suffer only 3 000 strokes. The difference is clearly seen. In the light of these figures it is reasonable to wish that, by hardening or case-carburising the tyre, or by adjusting its composition, it were possible to confer upon its running surface the greatest possible wear resistance consistent with freedom from brittleness, economical price and the possibility of subsequent restoration of profile by machining.

II. — The use of a transparent tracing facilitates the study of the proposed tyre profile with special regard to the *location of maximum wear*. It may be distributed over the entire height, as with steep flanges, at the risk of introducing severe rubbing contact well in advance of the point of support; or placed midway only as with certain rounded flanges used in the past; or at the root as in the case of the receding flanges at present in use.

Note on working stresses. Do these promote the establishment of a common profile of equilibrium?

Let us put aside the question of wear and consider the expression derived by HERTZ for the maximum local stress q in two elastic bodies that are in contact. For steel, with a modulus of elasticity of 2.10^{10} this may be written :

$$q = \frac{0\ 2444\ P^{1/3}E^{2/3}S^{2/3}}{mn} = \frac{1.80\ P^{1/3}S^{2/3}}{mn}\ 10^6$$

Where P is the component of the weight of the wheel normal to the plane of contact;

S is the sum of the two principal curvatures of the rail profile and the tyre profile respectively;

m, n, are two elliptic functions selected from a table giving them in terms of the difference of the curvatures.

We have seen in the calculation of the meridian that the points capable of coming into contact, i. e. points having parallel tangents, have radii r_m and r_a which differ from each other only by the quantity

$$\frac{Ri^2}{\cos^3 t},$$

where t is the inclination of the rail profile to the horizontal at the point of contact. If, starting from this we calculate S we find, for the two principal radii of curvature and their two inverse radii (fig. 10) that the sum of the curvatures is

$$S = \frac{1}{r_a} - \frac{1}{r_a + \frac{Ri^2}{\cos^3 t}} + \frac{\cos t}{R}$$

	Rail *cylindrical surface*		*Tyre* *running surface*	
Radii of curvature....	r_a	infinity	$-(r_a + \frac{Ri^2}{\cos^3 t})$	$\frac{R}{\cos t}$ Th. of Meusnier
Curvatures.....	$\frac{1}{r_a}$	0	$\frac{-1}{r_a + \frac{Ri^2}{\cos^3 t}}$	$\frac{\cos t}{R}$

For simplicity, we will only consider the angle $t = 45^0$, in which case $\cos t = 0.707$ and $\cos^3 t = 0.35$. Taking the standard

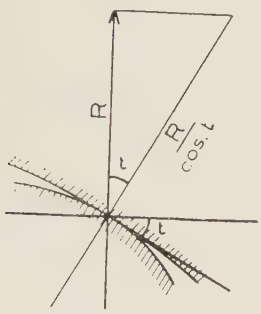


Fig. 10.

46 kgr. rail when new, then $r_a = 13$ mm. For a wheel 1 metre in diameter R = 0.500 and $Ri^2 = 1$ mm. (mean value) whence :

$$S = \frac{10^3}{13} - \frac{10^3}{13 + \frac{1}{0.35}} + \frac{0.707}{0.500}$$

$$= \frac{10^3}{13} - \frac{10^3}{13 + 2.85} + 1.41 = 15$$

From the table $1.80\,mn = 0.711$ and the stress in the metal for a wheel loaded whith 6 metric tons is:

$$q = \frac{6\ 000^{1/3} 15^{2/3}}{0.711} \cdot 10^6 = \frac{20 + 6.08}{0.711}\ 10^6$$

= 171 kgr./mm² (108.5 tons/ sq. inch).

Although this stress may appear high, it is quite normal in a problem of this nature where one is considering extremely small contact surfaces encircled by restraining metal.

If it had been assumed that the rail profile were worn to a radius of 8 mm. the same calculation would have shown that $q =$ 249 kgr./mm² (158 tons/sq. inch.). Such a stress will promote rapid breakdown of the metal and the probable early return to a larger radius.

Experience alone will show whether there exists a radius at which rail wear may be stabilised.

The assumption that the mutual wear of rail and tyre may result in an *accurately predictable mode of wear* and, as experience suggests, in diminution of the corner radius, leads one to *speculate whether*, in order to achieve a certain economy of material, the wear *might be anticipated.* The experiment might be made (see the Conclusion) but *a priori* we think that it will fail and we consider that there is more to be gained by providing a large radius at the wearing face, such as the 13 or 15 mm. of the present day. In this way the corner of the rail is much less heavily loaded and the disintegration of the metal correspondingly reduced.

The sharp flange. — There appears little likelihood of any close agreement being reached on the definition of the sharp or rubbing flange.

Let us imagine that a tyre, as a result of running conditions, has worn to a rather steep profile resembling one of the Ri^2 curves that we are now studying. There may be a pronounced ridge at the junction between the worn surface and the unaffected portion of the flange. The ridge will be blunt, but clearly defined. Suppose further that this tyre now runs over a route where the angle of incidence i is much accentuated as the result, say, of the radius of curvature or of the track being wide to gauge. The ridge of the flange may bite into the rail at a point of second contact well ahead of the point of support of the wheel. At this point the relative displacement will be high, resulting in great friction and harsh contact; in brief, all the elements of abrasive contact. If this is a cutting flange then it would appear to consist of a sharply ridged profile too steep for the particular track on which it is made to run.

It could only be considered too sharp for certain kinds of track. The requirements for locomotives and coaches running continually over certain favourable main line routes might be quite different from those for wagons that have to run over all lines, even the most sinuous, or for stock and locomotives destined for service on sharply curving track. All the foregoing do, however, have to run over identically similar subsidiary track and fittings and will perhaps conform thereby to a common law.

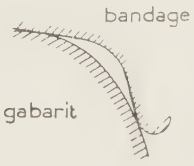


Fig. 11.

Bandage = tyre. — Gabarit = template.

The *way to judge a rubbing flange* would be to apply to the tyre a *template shaped to the contour of a meridian* Ri^2 with which it must not come into second contact in a zone that is too steep or too far removed from the running plane (fig. 11).

No attempt will be made here to lay down any value for Ri^2 that might be taken as an invariable characteristic of a rubbing flange. We appeal to the experience of all our colleagues for a solution of this question.

Conclusion.

Notes and suggested experiments. — We do not suppose that the present study can lead forthwith to firm conclusions or, above all, to numerical solutions, which would be premature. At the best the conclusions to which it may eventually lead would not appear to be revolutionary; but they will doubtless confirm the sound basis for the choice of the receding flanges in current use. (See the *Congress Bulletin* for November 1932, p. 2170, which shows all the established forms of tyre profile.)

The study does however offer a simple approach and indicates the need for a certain number of practical trials and experimental observations.

(1) It is first of all *desirable that the angles of incidence* of different vehicles on a variety of curves *should be observed and measured.* It would seem that direct recordings employing devices analogous to those so ingeniously conceived by M. MAUZIN for the former Orleans system, should to-day be practicable. (See *Revue Générale des Chemins de Fer*, January 1933.)

(2) It is desirable to measure *the progressive wear of a tyre* running continually under the same conditions. Are the last few square millimetres of cross-section to be worn away removed much more or much less rapidly than the first?

To ascertain this it would be necessary to use a known vehicle behind which one would periodically travel, such as a locomotive attached to one depot, a passenger vehicle

running always on the same route, a wagon reserved for service on one local route, or a tram. It would be of great interest to observe the successive modes of wear and their progression.

(3) Finally, a wealthy organisation such as a group of American railroads or a nationalised European system, *could conduct multiple trials* of the behaviour of different tyre profiles in the presence of varying rail profiles.

We envisage for instance a circular track of 600 metres radius within which there would be tracks of 150 metres radius (as for subsidiary track) 300 metres (as in branch lines) and perhaps also the intermediate radius of 225 metres encountered in mountain lines used by electric locomotives running on bogies.

The smaller tracks would be arranged within the large one either concentrically or otherwise.

Vehicles of chosen wheelbase would run on them either singly or in identical rakes. The play of each axle relative to the frame would be previously determined. Selected tyre profiles would be periodically checked and the wear estimated after successive trips of a certain number of kilometres. It would then be possible to see whether certain parts of the tyre section wear more quickly than others.

In order to expedite the measurements and minimise the cost of the trials, the tyres might be made of *softer material*, such as mild steel or an even softer metal. Careful comparisons would show whether the results so obtained were valid or whether they were falsified by the effects of cold work (*).

(*) There is the possibility that successful laboratory experiments might be made with one wheel only and a very short rail either in the form of a circle of a few metres diameter revolving beneath the wheel (?) or a straight length moving to and fro.

While the foregoing observations were in progress it would be necessary from time to time to determine the *angle* of incidence by means of recording apparatus. Its variation with type of vehicle, speed and wear could be assessed.

These trials would enable one to evolve *a tyre profile offering the greatest economy* from the standpoint of wear and renewal.

Moreover, by laying the curves first *with one type of rail*, and later with others, the influence of rail profile upon tyre wear would be apparent.

We do not think that any significant inferences could be drawn regarding the *wear of the rail* itself, since even with the 150 metre radius (940 metres circumference) which would be the shortest, each point on the rail would only suffer the contact of two wheel pairs per vehicle for each complete circuit, whereas each tyre type of one metre diameter makes 300 contacts. It would however be possible to roll a soft rail for the purpose of the trial. The cost would be high and partly irrecoverable. An appreciable amount would be lost by wear.

(4) It is possible that the method of envelopes and meridians, supplemented by the experimental results, would lead to new knowledge concerning *trams running on grooved rails*. These present a problem which appears so far to have received less attention, on account of its difficulties, than that of ordinary rails.

Track trials would be particularly simple, because the radii are so much smaller. A circular track 15 or 18 metres in diameter would only represent a total length of 100 metres and a figure-of-eight track no more than 200.

Experiments with solft tyres would quickly show the rates of wear of the different elements of the tyre facing the rail and the guard-rail.

APPENDIX I

THE RAIL. DETERMINATION OF THE ENVELOPE MERIDIAN

Figure 12 is a diagram representing the axis (EF, E' F') of the axle, and showing :

i the obliquity, always slight, of the rail with respect to the plane of the wheel, or angle of incidence;

RC, R' C' the section of the head of the rail in the vertical plane of the axle. This for practical purposes is the same as the normal cross section;

A, A', points in the rail profile of which *x*, *y* are the co-ordinates with respect to the axes *o x y'* of the figure.

It is evident that A' B' generates a hyperboloid with its hyperbolic meridian H' A' J' centred upon the axle, its apex at A and asymptotes whose angular coefficients are $\pm i$. It can be shown that the equation to the meridian is :

$$(H) \equiv Y^2 - y^2 - \frac{1}{i^2}(X - x)^2 = 0 \quad . \quad (1)$$

The envelope for these hyperbolae is obtained by differentiating the foregoing equation with respect to the parameters *(x y)* which are themselves related by the equation to the rail profile, and eliminating *x* and *y* from the three equations.

If the equation to the rail profile be

$$y = f(x)$$

the envelope will be defined by the three equations.

$$H \equiv Y^2 - y^2 - \frac{1}{i^2}(X - x)^2 = 0 \quad . \quad (1)$$

$$\frac{dH}{dx} \equiv -2yf'x + \frac{2}{i^2}(X - x) = 0 \quad . \quad . \quad (2)$$

$$y = f(x) \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

from which we obtain

$$X - x = i^2 y f' x \quad . \quad . \quad . \quad (2)'$$

Introducing *t*, the angular coefficient of the tangent to the rail profile

$$X = x + t i^2 y \quad . \quad . \quad . \quad . \quad . \quad (4)$$

whence $Y^2 - y^2 = t^2 i^2 y^2$

and $Y = y\sqrt{1 + t^2 i^2} \quad . \quad . \quad . \quad . \quad (5)$

For any given profile of rail head, radius of wheel, and obliquity *i*, equations (4) and (5) permit the construction point by point of the profile of the envelope.

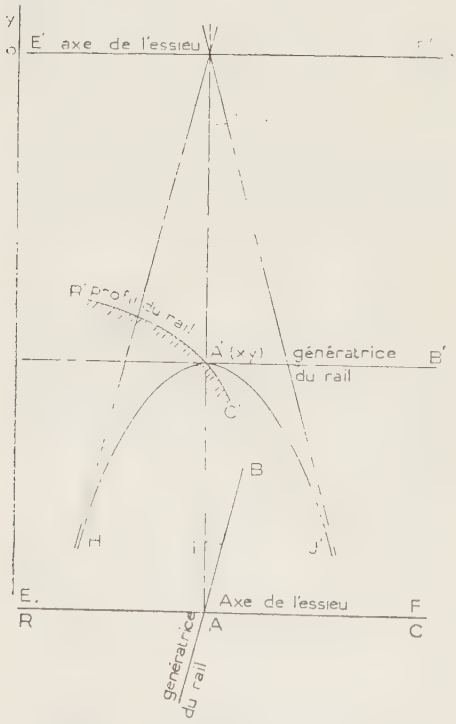


Fig. 12.

Explanation of French terms :

E' axe de l'essieu = centreline of axle. — R' profil du rail = profile of rail. — Génératrice du rail = generatrix of rail.

The meridian is obtained by a simple geometrical construction :

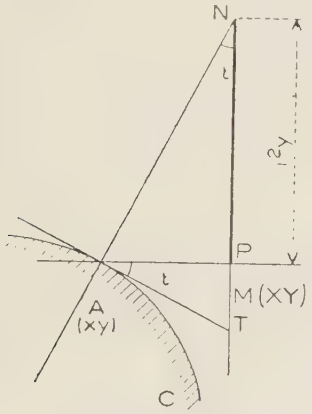
(1) **Abscissa.** — In figure 13 A C is the profile of the rail, A N is the normal at A and M the point (X Y) on the envelope derived from A *(x y)* on the head of the rail. In the figure :

$$A\,P = X - x = t\,i^2\,y$$

But $A\,P = t.\,N\,P$

and so $N\,P = \frac{A\,P}{t} = i^2\,y$

In this expression i^2 is constant and y alone varies with the point A. It will suffice therefore for each point A to set up a vertical $i^2 y$ between the normal A N and the horizontal A H in order to obtain the abscissa for the corresponding point M.



Fundamental construction.

Fig. 13.

The value of y varies very little over the active height of the rail head, which is less than 15 mm. in practice. It is therefore permissible to assign to N P the *constant value* Ri^2, where R is the effective tread radius of the wheel, or this value augmented by from 0 to 15 mm.

(2) **Ordinate.** — Further P M = Y — y. We may then write :

$$Y^2 - y^2 = t^2 i^2 y^2 = (Y - y)(Y + y) \quad . \quad (1)'$$

$$\text{or } Y - y = \frac{t^2 i^2 y^2}{Y + y} = \frac{(X - x)\, t\, y}{Y + y}$$

$$\text{or } \frac{Y - y}{X - x} = \frac{t\ y}{Y + y} = t \frac{y}{2y + MP} = \frac{t}{2} \frac{1}{1 + \frac{MP}{2y}}$$

It is permissible to neglect M P in relation to the diameter 2 y of the wheel within a zone equal to the height of the flange under consideration, in which case :

$$\frac{Y - y}{X - x} = \frac{t}{2}$$

and *the point M on the envelope lies in the middle of the segment T P.* To be strictly accurate, it should lie closer to P by an amount between 0 and 3 % of the value of P, and the curve that we are to trace will in consequence be slightly too steep.

(3) **Tangent.** — The value u of the angular coefficient of the envelope at M may be calculated from equations (4) and (5), its exact value being :

$$u = \frac{t}{\sqrt{1 + t^2 i^2}} \quad . \quad . \quad . \quad . \quad . \quad (6)$$

Since i is small, about 1/10 to 1/20, $t^2\ i^2$ is very small within the limits studied, and we find therefore that $u = t$. This means that the *tangents at points A and M on the rail and the envelope respectively are parallel* over that portion of the meridian lying nearest to the horizontal. For the highest values of t, u diminishes slightly and here again the drawn curve will be a little steep.

In brief, the *construction* gives a very close approximation to the *envelope*. It is a *simple anamorphosis of the profile of the rail* and is obtained by the three following operations :

— trace the normal at each point A on the rail profile and set out the subnormal P N = Ri^2, which is constant for a given value of i. (R is the wheel radius plus from 0 to 15 mm. and i the obliquity of the axle with respect to the rail);

— on the vertical N P lay off P M = T P. M belongs to the envelope;

— the tangent to the envelope at M is parallel or very nearly so to the tangent to the rail at A.

This simple construction requires a rule and a set square only and no compasses. It is much quicker than any so far employed in the study of this problem.

Remarks.

In the foregoing a number of approximations have been permitted, but a rigorously exact geometrical solution is available.

(1) **Abscissa.** — The precise value for the abscissa is obtained by taking $i^2 y$ as the subnormal throughout instead of the approximate value Ri^2.

(2) **Ordinate.** — In the place of the approximation M P = A P $\frac{t}{2}$, the exact value must be used :

$$MP = AP \frac{t}{2} \frac{2y}{2y + MP}$$

The first value of M P may be corrected by reducing it in the radio $\frac{2y}{2y + MP}$, either by the use of similar triangles or by calculation. The latter is more convenient on account of the smallness of M P by comparison with y. For instance, if M P is found to be 20 mm. and $2y = 1\,005$ mm., the former is multiplied by $\frac{1\,005}{1\,025}$, i. e. it is reduced by $\frac{2}{100}$ or 0.4 mm.

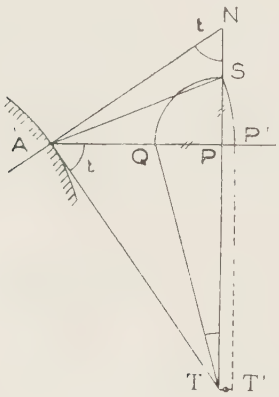


Fig. 14.

(3) **Tangent.** — The exact value of the tangent is :

$$u = \frac{t}{\sqrt{1 + t^2 i^2}}$$

Draw the angle P T Q = i (fig. 14). Then P Q = T P. i = A P. t. i. With radius P Q mark off P S. Then

$$AS = \sqrt{\overline{AP}^2 + \overline{PS}^2} = AP\sqrt{1 + t^2 i^2}$$

With radius A S mark off AP' and translate the point T to T'. Calculating the angular coefficient of the straight line AT' we have :

$$\frac{T'P'}{AP'} = \frac{TP}{AP}\frac{AP}{AP'} = \frac{TP}{AP}\frac{AP}{AS} = \frac{t}{\sqrt{1 + t^2 i^2}}$$

It will be noted that :

(1) If *the rail* is *worn to a sharp corner* A a part of the envelope will consist of the hyperboloid generated by the sharp ridge A.

The corresponding part of the meridian would then be a hyperbola having axes of length R and Ri, a radius of curvature Ri^2 at its apex and asymptotes of obliquity $\pm i$.

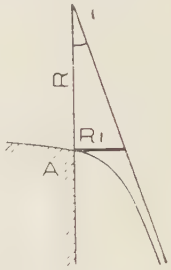


Fig. 15.

(2) If the *rail* is bevelled at *an angle* t to the *running plane* (see fig. 16) the corresponding part of the envelope is a cone and its meridian a

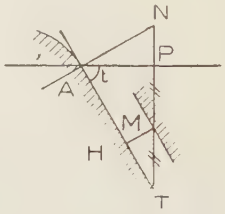


Fig. 16.

straight line parallel to the bevelled cheek of the rail and removed from it by a distance $\frac{Ri^2 \sin^2 t \, tg \, t}{2}$ which is very nearly the same as $\frac{1}{2}$ Ri^2 tg t for large values of t. (If for example tg t = 4, the distance becomes 2 Ri^2.)

APPENDIX II

RADIUS OF CURVATURE OF THE MERIDIAN OR TRANSFORMED CURVE OF THE RAIL PROFILE.

If x be the abscissa of the point A and z that of the homologous point M, then putting K for the product Ri^2 :

$$AP = K \text{ tg } t$$

and

$$z - x = K \text{ tg } t$$

whence

$$dz = dx + \frac{K}{\cos^2 t} dt \quad . \quad . \quad . \quad . \quad (1)$$

The arcs of the two curves A and M being ds and dm, and the tangents being parallel, we have $dx = ds \cos t$ and $dz = dm \cos t$.

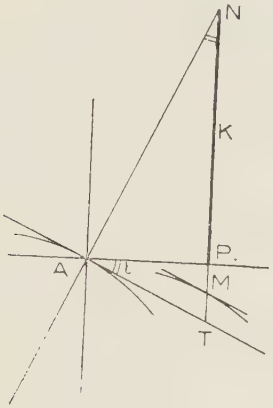


Fig. 17.

But $\frac{ds}{dt}$ and $\frac{dm}{dt}$ are the radii of curvature r_a and r_m of the two curves. Thus, dividing both variables of equation (1) above by $\cos t \, dt$ we obtain :

$$\frac{dz}{\cos t \, dt} = \frac{dx}{\cos t \, dt} + \frac{K}{\cos^3 t}$$

or $$\frac{dm}{dt} = \frac{ds}{dt} + \frac{K}{\cos^3 t}$$

or $$r_m = r_a + \frac{K}{\cos^3 t} = r_a + \frac{R i^2}{\cos^3 t} \quad . \; . \; . \quad (2)$$

Example : For the middle portion of the corner radius, where $t = 45^\circ$ approx : $\cos t = 0.707$ and $\cos^3 t = 0.35$. The difference of the two radii of curvature becomes $\frac{\cos^3 t}{R i^2}$ or 2.85 $R i^2$.

Where $R i^2 =$	0.5	1.0	1.5	2.0 mm.
$r_m - r_a =$	1.4	2.8	4.3	5.7 mm.

If the corner radius of the rail is 13 mm., then

$r_m =$ 14.4 15.8 17.3 18.7 mm.

APPENDIX III

EQUATION TO THE TRANSFORMED CURVE

The co-ordinates of the point A being x and y, the co-ordinates of the point M will be

$$x_1 = x + A\,P = x + K \operatorname{tg} t = x + x_2$$

$$y_1 = y - P\,M = y - \frac{K}{2} \operatorname{tg}^2 t = y - y_2$$

$$\text{where} \begin{cases} x_2 = K \operatorname{tg} t \\ y_2 = -\frac{K}{2} \operatorname{tg}^2 t \end{cases}$$

It is seen that the co-ordinates of the point M of the transformed curve are obtained from the co-ordinates of the rail by the addition of the elements x_2 and y_2 which are the co-ordinates of a parabola having a vertical axis, since

$$x_2^2 + 2 K y_2 = 0$$

To the co-ordinates of the point A on the rail are added those of a point on the parabola having the same angular coefficient.

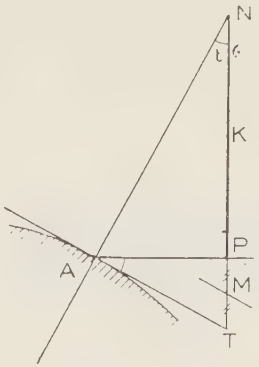
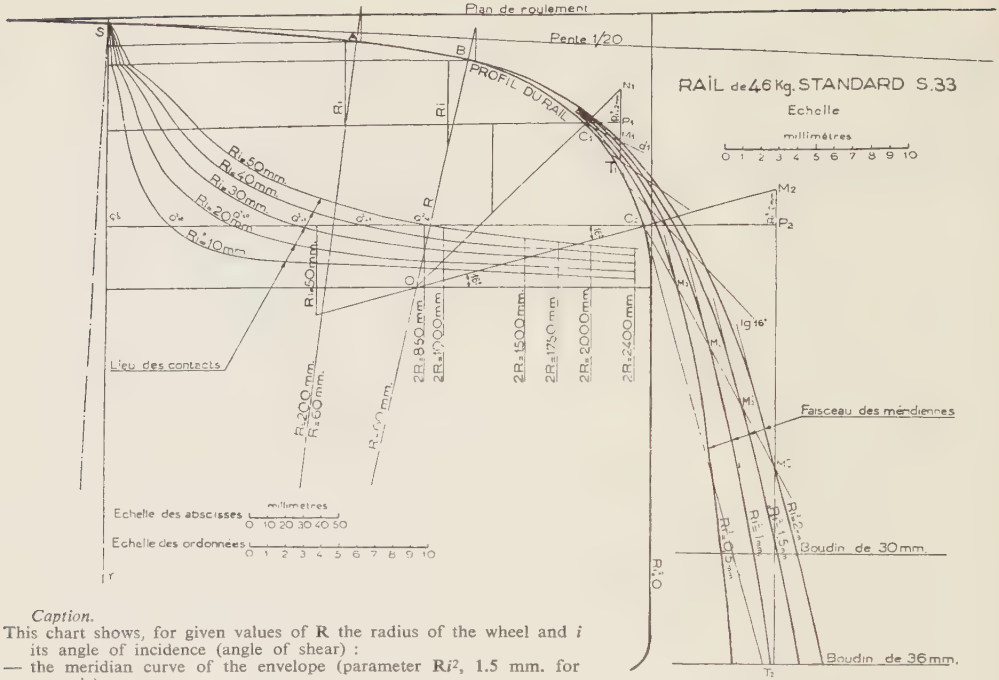


Fig. 18.

See *Appendix IV* pages 532 and 533.

PLATE I. **Standard 46 kgr. Rail. S. 33.**


Plan de roulement
Pente 1/20
PROFIL DU RAIL
RAIL de 46 Kg. STANDARD S.33
Echelle
millimètres
0 1 2 3 4 5 6 7 8 9 10
Lieu des contacts
Faisceau des méridiennes
Boudin de 30 mm.
Boudin de 36 mm.
Echelle des abscisses
millimètres
0 10 20 30 40 50
Echelle des ordonnées
0 1 2 3 4 5 6 7 8 9 10

Caption.

This chart shows, for given values of R the radius of the wheel and i its angle of incidence (angle of shear) :

— the meridian curve of the envelope (parameter Ri^2, 1.5 mm. for example);

— the second contact, such as M'''_2, between the transparent tracing of the tyre profile and the meridian;

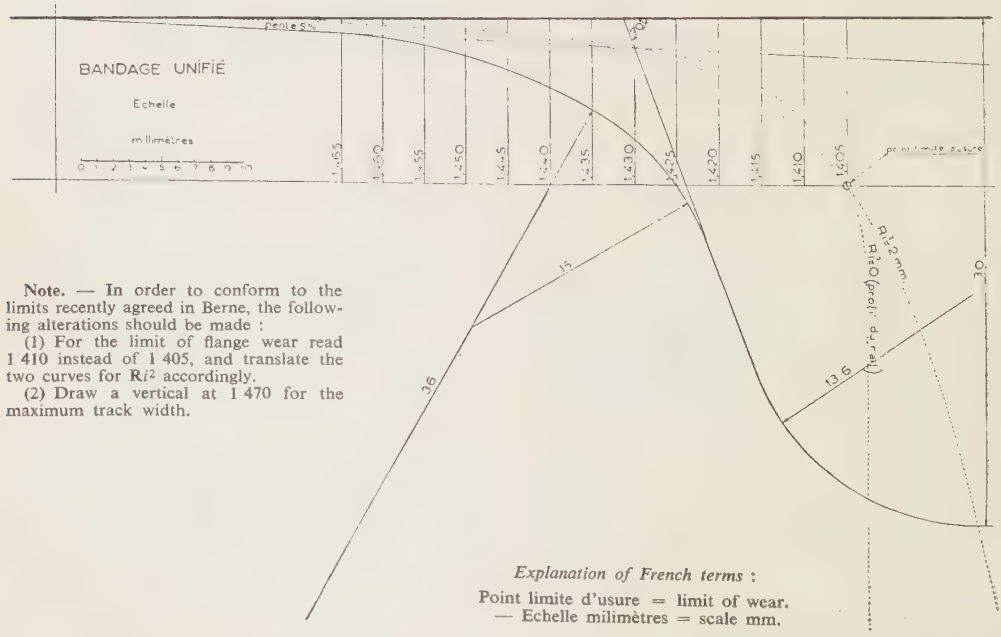
— the generatrix C_2P_2 of the rail profile along which second contact takes place and the inclination C_2T_2 of the tangent plane;

— the curve for the locus of second contact (parameter say Ri = 40 mm.) and the real abscissa $Q^2_0\ Q^2_{40}$ corresponding to 145 mm. of second contact.

Explanation of French terms :

Plan de roulement = running plane. — Pente 1/20 = slope 1/20. — Profil du rail = profile of rail. — Échelle = scale. — Lieu des contacts = locus of contacts. — Faisceau des méridiennes = family of meridians. — Échelle des abscisses = scale of abscissae. — Échelle des coordonnées = scale of coordinates. — Boudin = flange.

PLATE II. **Existing tyre profile.**



Note. — In order to conform to the limits recently agreed in Berne, the following alterations should be made :

(1) For the limit of flange wear read 1 410 instead of 1 405, and translate the two curves for Ri^2 accordingly.

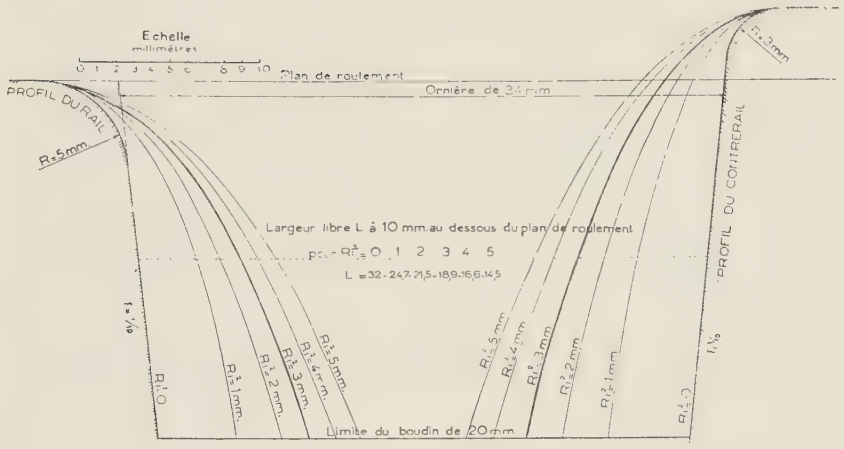
(2) Draw a vertical at 1 470 for the maximum track width.

Explanation of French terms :

Point limite d'usure = limit of wear.
— Echelle milimètres = scale mm.

PLATE III.

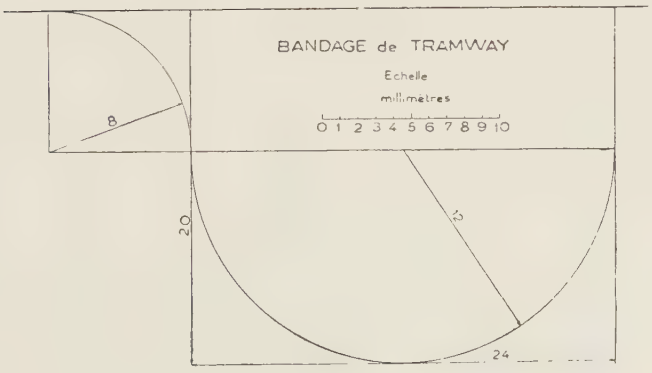
Rail with sunken flangeway IC2.



Explanation of French terms :

Ornière de 34 mm. = flangeway 34 mm. — Profil du rail = profile of rail. — Profil du contrerail = profile of checkrail. — Largeur libre L à 10 mm. au-dessous du plan de roulement = free width L at depth of 10 mm. below running plane. — Limite du boudin de 20 mm. = limit for 20 mm. flange.

PLATE IV.



Explanation of French terms :

Bandage de tramway = Tramway tyre.

APPENDIX IV

PROVISIONAL TABLE OF ANGLES OF INCIDENCE

Type of vehicle	*Standard track*											
	short bogie		*long coach. two axles*		*loco: 6 ft. 6 inch. wheels*		*long railcar, parallel axles*		*Metro bogie*		*long tram*	
Condition of vehicle and of track ⟶	*new*	*worn*	*new*	*worn*	*new*	*worn*	*new*	*worn*	*new*	*worn*	*new*	*worn*
Width of track ..... mm.	1 435	1 470	1 435	1 470	1 435	1 470	1 435	1 470	1 440	1 445	1 438	1 445
Total width over flanges ..... mm.	1 426	1 410	1 426	1 410	1 426	1 410	1 426	1 410	1 425	1 400	1 432	1 423
Play j = ... mm.	10	60	10	60	10	60	10	60	10	55	6	22
Wheelbase L ..... m.	2.500		9.250		6.150		9.800		1.800		3.400	
Play of axles parallel to track j'= ..... mm.	11	29	11	29	1	1	1	29	4	10	2	10
Width between horns e' ..... m.	1.810		1.810		1.190		1.810		1.752		1.700	
End play of axles ..... mm.	12	41	12	41	3	6	12	41	5.5	12.5	10	25
Radius of curvature C_1 ..... m.				250								
Radius of curvature C_2 ..... m.				150					100			
Radius of curvature C_3 (local lines only) m.				50					75		18	
Obliquity $\frac{j+j''}{L}$ = ... $10^{-3}\times$	8.4	40.4	2.3	10.9	2.0	10.7	2.1	10.3	8.6	37.5	4.7	13.8
Obliquity $\frac{j'}{e'}$ = ... $10^{-3}\times$	5.8	16	5.8	16	0.8	0.8	0.5	16	2.3	5.7	1.2	5.9
Obliquity $\frac{L}{2C}$ for C_1 ..... $10^{-3}\times$	5		18.5		12.3		19.6					
Obliquity $\frac{L}{2C}$ for C_2 ..... $10^{-3}\times$	8		30.8		20.5		33		9			
Obliquity $\frac{L}{2C}$ for C_3 ..... $10^{-3}\times$	25				(61.5)		(98)		12		94.5	
Total obliquity i for C_1 ..... $10^{-3}\times$	18.9	61.4	26.6	45.4	15.1	23.8	23.2	45.9				
Total obliquity i for C_2 ..... $10^{-3}\times$	21.9	64.4	38.9	57.7	23.3	32.0	36.6	59.3	22.9	52.2		
Total obliquity i for C_3 ..... $10^{-3}\times$	38.9	81.4							25.3	55.2	100.4	114 %
Diameter of wheel ..... mm.	1 050		1 050		2 000		1 050		850		830	
Radius of wheel ..... mm.	525		525		1 000		525		425		415	
Value of Ri ..... mm.	10 to 43		14 to 30		15 to 33		12 to 31		8 to 23		43	47
Value of Ri^2 ..... mm.	0.19 to 3.5		0.4 to 1.8		0.2 to 1.0		0.28 to 1.8		0.17 to 1.3		4.2	5.4

PROVISIONAL TABLE OF ANGLES OF INCIDENCE *(Continued.)*

Type of vehicle	*Metre gauge track*						*60 cm. track*			
	bogie		*vehicle, 3 metres wheelbase*		*locos: 4 ft. 7 inch. wheels*		*short bogie*		*long loco*	
Condition of vehicle and of track ——→	*new*	*worn*	*new*	*worn*	*new*	*worn*	*new*	*worn*	*new*	*worn*
Width of track ..... mm.	1 000	1 015	1 000	1 015	1 000	1 015	600	615	600	615
Total width over flanges	990	970	990	970	990	970	590	580	590	580
Play j = ... mm.	10	45	10	45	10	45	10	35	10	35
Wheelbase L ..... m.	1.20		3		1.80		0.60		1.40	
Play of axles parallel to track j'= ..... mm.	8	15	8	15	1	1	5	12	0.5	3
Width between horns e' = ... m.	1.40		1.40		1.40		0.90		0.456	
End play of axles ..... mm.	8	25	8	25	2	4	5	15	2	5
Radius of curvature C_1 ..... m.			100					30		
Radius of curvature C_2 ..... m.			50							
Radius of curvature C_3 (local lines only) m.			30					20		
Obliquity $\frac{j + j''}{L}$ = ... $10^{-3}\times$	15	58	6	23	6.7	27	25	83	8.6	28
Obliquity $\frac{j'}{e'}$ = ... $10^{-3}\times$	6	11	6	11	0.7	0.7	5.5	13.3	1.1	6.6
Obliquity $\frac{L}{2C}$ for C_1 ..... $10^{-3}\times$	6		15		9					
Obliquity $\frac{L}{2C}$ for C_2 ..... $10^{-3}\times$	12		30		18		10		23	
Obliquity $\frac{L}{2C}$ for C_3 ..... $10^{-3}\times$	20		50		30		15		35	
Total obliquity i for C_1 ..... $10^{-3}\times$	27	75	27	49	16.4	36.7				
Total obliquity i for C_2 ..... $10^{-3}\times$	33	81	42	64	25.4	45.7	40.5	106.3	32.7	57.6
Total obliquity i for C_3 ..... $10^{-3}\times$	41	89	62	84	37.4	57.7	45.5	111.3	44.7	69.6
Diameter of wheel ..... mm.	720		850		1 400		400		600	
Radius of wheel ..... mm.	360		425		700		200		300	
Value of Ri ..... mm.	9.3 to 32		11.5 to 36		11.5 to 40		8.1 to 23.3		9.8 to 20.8	
Value of Ri^2 ..... mm.	0.27 to 2.8		0.32 to 3		0.19 to 2.3		0.33 to 2.5		0.3 to 1.4	

[621 .132 .3 (496)]

Turkish State Railways, 2-10-0 Locomotives.

(The Locomotive, December 15, 1948.)

Thirty-seven 4 ft. 8 1/2 in. gauge 2-10-0 locomotives have recently been delivered by Beyer, Peacock & Co. Ltd. and the Vulcan Foundry Ltd. to the Turkish State Railways. These engines, built to the inspection of Robert W. Hunt Co., were ordered in 1939 but owing to the war could not be completed until this year. The design is similar to that of the « 1E » class, previously built by Henschel & Son of Kassel and Krupp of Essen. The engines were shipped fully erected; due to their large size they had to be conveyed by road to the docks at Liverpool and were shipped to the Turkish port of Haydar Pasha.

Boiler of Turkish 2-10-0 locomotive.

The large boilers, with the centre line 10 ft. 2 in. above rail level, have a diameter at the front end of 6 ft., the second ring being 5 ft. 10 7/8 in. diameter. The barrel is exceptionally long being 19 ft. 8 1/4 in. between tubeplates; overall with smokebox the length of the boiler is 38 ft. 5 1/2 in. and the weight empty 31 tons. The round-top firebox has a copper inner-box, the water space stays being of the same material. The steel roof-stays are nutted on the inner side but the ends are drilled taper and expanded with a drift to give a steam tight fit without riveting. The cross stays are screwed into seatings and welded.

The tubes are steel, expanded into the tubeplate and beaded at the firebox end. There are 106 of 2 1/8 in. and 43 of 5 5/8 in. outside diameter, they contribute 2 406 sq. ft. to the total of the 2 575 sq. ft. of evaporative heating surface — the firebox accounting for 169 sq. ft. The grate area is 43 sq. ft. and the working pressure 228 lb. sq. in.

The superheater, manufactured by the Superheater Co. Ltd., is of special design, the saturated and superheater header being cast in two parts. These when bolted together have the saturated and superheated seating alternating. The superheater has forty-three elements and an area of 908 sq. ft.

The ashpan is of the three hopper type, the discharge doors being operated from the cab. The grate has four sets of bars, three of these

are fixed the remaining set being arranged to drop.

A No. 11 exhaust injector, type « H » and a No. 11 live-steam injector, type « ASZ », both by Davies & Metcalfe, deliver feedwater to the purifier housed in the front dome. The purifier is in the form of cascade trays and the water is delivered from the clack boxes, via twin opposed nozzles, in the form of a spray. There is a connection on the injector delivery pipe for the attachment of a firehose—a feature sometimes met with in American practice.

2-10-0 Locomotive. Turkish State Railways.

An « Everlasting » blow-off cock, operated from the cab, is mounted on the firebox front.

The frames, machined from rolled-steel slabs, are of the bar type 3 1/2 in. thick. The suspension is compensated, the leading truck and first and second coupled axles on one side forming one group and the third, fourth and fifth coupled axles constituting the second group per side.

The axleboxes are of forged steel, fitted with white-metalled loose brasses; the driving boxes have additional segments below the centre-line fitted with shims to allow vertical adjustment. The leading and trailing axleboxes have a lateral movement of 1 3/16 in. each way. The Krauss truck has 4 7/8 in. movement on each side.

The leading truck wheels are of 2 ft. 9 1/2 in. diameter, that of the coupled wheels being 4 ft. 9 1/8 in.

The cast-iron cylinders are 25 5/8 in. bore by 26 in. stroke and in addition to being bolted to the frames are keyed by fitting strips. Three narrow rings are fitted on the forged steel pistons. Hollow tail rods are fitted, the guides having vertical adjustment. The 11 13/16 in. diameter piston valves have two cast-iron heads each with four narrow rings; the travel in full gear is 7 5/16 in. Large relief valves are fitted on the cast-steel covers and air-operated bye-pass valves are mounted on the top of the cylinders.

Straight air and automatic Westinghouse brake equipment is fitted, air being supplied by a 7 in. cross-compound compressor. Riggenbach counterpressure apparatus is provided for use when descending long grades with the engine reversed. A rotary valve in the exhaust pipe, operated by a small air cylinder, closes the exhaust outlet and opens a port, situated on the smokebox side, to atmosphere. This enables clean air, which is mixed with water spray as a coolant, to be drawn into each cylinder. After being com-

pressed in the cylinders the air passes into the main steam pipes and is exhausted, via a cab-controlled needle valve, behind the chimney.

The cab lay-out generally follows the practice of the German State Railways. Amongst items of equipment may be mentioned the Wakefield mechanical lubricator, driven from the left-hand trailing crankpin. This fourteen feed lubricator is situated in the cab and has mounted above it a separate sight-feed indicator. All feeds are provided with « Olva » diaphragm check valves manufactured by Davies & Metcalfe Ltd. The Marcotty type firehole door has a hollow cast steel frame admitting heated secondary air between two spaced steel plates.

The tender runs on two diamond framed bogies with 3 ft. 3 $^{3}/_{8}$ in. diameter wheels. The coal capacity is 7 tons 17 cwt. and 6 380 gallons of water are carried. The tenders built by Beyer, Peacock & Co. Ltd. are riveted whilst those constructed by the Vulcan Foundry Ltd. are welded.

The engine weight in working order is 106 tons 18 cwt., the riveted tender turns the scale at 61 tons 19 cwt. whilst the welded one weighs 60 tons 16 cwt.

Calculated at 85 per cent. the tractive effort is 57 560 lb. giving a factor of adhesion of 3.64.

René HENNING.

[385. (09.2]

OBITUARY.

René HENNING,

Honorary General Manager of the Belgian National Railways.

Former President of the Permanent Commission of the International Railway Congress Association.

M. René Henning, Former President of the Permanent Commission of our Association, Honorary General Manager of the Belgian National Railways, and Vice-President of the Administrative Council of the Luxembourg Railways died at Bagnoles-de-l'Orne on the 3rd September 1948.

It will be remembered that the International Railway Congress Association resumed its activities after the calamities of 1940-1945 under the presidency of M. Henning.

M. Henning was unanimously nominated as President of the Permanent Commission of our Association at the first meeting of the Commission held after the war, and it was owing to him that the invitation from the Swiss Government to hold a Congress at Lucerne in 1947 was accepted and the organisation of this first post-war Congress was commenced and actively pursued.

M. Henning had long been a devoted collaborator of the Association, his collaboration in our work dating back to 1933 when he was Vice-President of Section IV : General, at the Cairo Congress.

In addition, he published several very noteworthy articles in the monthly *Bulletin*, particularly in connection with railway rating policy and the co-ordination of transport.

Born in Brussels on the 20th August 1881, M. René Henning had a brilliant record as a student of modern humanities, which he interrupted at the University stage to enter the employment of the Belgian State Railways at the age of 18. Although entrusted with much absorbing work, he continued to improve his classical education by the study of the classical languages.

During the 1914-18 war, as the State Railways were not working, M. Henning profited by his enforced leisure to study law, in particular those branches concerned with contracts, and in particular transport contracts.

He put his new knowledge at the service of the Railways, and to their profit made a brilliant success of a series of litigation concerning the revision of long term contracts, on account of want of foresight.

He also published in collaboration with M. Eugène Stevens, the lawyer, a treatise on *Transport Contracts* which ranks high in legal literature.

When the Belgian National Railways were created, he was nominated Assistant to the Operating Manager.

His international career dates from this time. He took part in numerous conferences at which he always distinguished himself by his clear-sightedness and logic, so that thanks to his personal prestige Belgium occupied a very important place at these meetings.

During the last war, he was promoted Adviser to the General Manager.

On the 15th November 1944, he was appointed General Manager of the Belgian

National Railways. Everyone knows the condition in which he found the installations and stock after the long years of occupation, sabotage and bombardments suffered by the railway. He had however to assure the large scale military transport which made possible the victorious progress of the military operations. With the assistance of his immediate collaborators, he dealt with this overwhelming task with such clear-sightedness and verve that even the most critical were won over.

On the 1st September 1946, M. Henning retired after a long and brilliant career of nearly 47 years.

But this indefatigable fighter did not intend to enjoy the rest he had so richly deserved. He was nominated as the representative of the Belgian Government at the E. C. I. T. O. and also a member of the Administrative Council of the Luxembourg Railways, the Vice-President of which he became. He devoted himself to these tasks, and those who worked with him will remember his labours and guidance, which always had such a personal touch.

M. Henning's brilliant career was crowned by many distinctions, both Belgian and foreign, amongst which mention may be made of the Commander of the British Empire and the Medal of Freedom with Bronze Palm.

His competence, dynamic personality, faculty for hard work, clear-sightedness, vivacity, and discernment, all of which characteristics he possessed to a high degree, make the loss occasioned by his death all the greater.

We feel his loss very greatly and ask his family to accept our sincere sympathy.

The Executive Committee.

[385. (06 .111]

OFFICIAL INFORMATION

ISSUED BY THE

PERMANENT COMMISSION

OF THE

International Railway Congress Association.

Enlarged Meeting of the Permanent Commission at Lisbon (1st to 4th June, 1949.)

FINAL SUMMARIES adopted at the Enlarged Meeting of the Permanent Commission of the International Railway Congress Association (Lisbon, June 1949).

1st SECTION : WAY AND WORKS.

QUESTION I.

a) Mechanisation of the maintenance and renewal of the permanent way.

b) Recent improvements relating to reinforced concrete and prestressed concrete sleepers. Results obtained.

c) Recovery and strengthening of metal bridges that have reached the theoretical limit of safety.

Summaries.

a) *Mechanisation of the maintenance and renewal of permanent way.*

« 1. The use of mechanical plant is not « yet general in all countries but in certain « countries it has already passed the « experimental stage for renewal work « and in some cases even for maintenance.

« 2. The use of mechanical methods « undoubtedly presents economic advant- « ages in carrying out renewal work and « for removing and cleaning ballast. It is « equally advantageous for maintenance, « particularly that of sleepers and fasten- « ings, provided the work is of sufficient « extent to justify the employment of « special gangs.

« 3. In addition to economic advant- « ages, the use of mechanical equipment « also results in appreciable technical « advantages.

« The work is often more uniform, « better and more lasting, compared with

« manual work; the time taken for cer-
« tain operations is shortened which is of
« great importance when the line is only
« available for a short period; the work
« is less fatiguing for the staff. It is
« possible to introduce or develop further
« new maintenance methods such as
« grinding the rails, building up the ends
« of rails and crossings by welding, etc.

« 4. To obtain a good output from both
« men and machines on work where
« either mechanical plant or a large
« number of modern hand tools are used,
« it is most necessary to examine in the
« fullest detail possible the organisation
« of the work and its subdivision into
« well defined tasks, each one allocated to
« a particular gang.

« 5. The maximum output from mechan-
« ical plant for removing and cleaning
« ballast and for removing and relaying
« track can be obtained when long inter-
« vals between the passing of trains are
« available. This practice enables renewal
« work to be accelerated, reduces the
« delay in carrying out such works on a
« given section of line and in particular
« reduces the expenditure of all depart-
« ments, especially the cost of reduced
« speeds.

« 6. In view of the frankly evolutionary
« stage of the matter in most countries
« it is not possible to formulate any
« definite summaries regarding the type
« of mechanisation most to be recom-
« mended nor the types of machines most
« suited to the different operations.

« 7. In general the use of mechanical
« plant greatly improves conditions of
« work : it constitutes therefore a factor
« in social progress which is not to be
« neglected.

« Since fewer men are required and to a
« large extent heavy manual work avoided,
« mechanisation of renewals and mainten-
« ance of the permanent way facilitates
« the recruiting of staff for permanent
« way and contract works, especially in
« industrial areas.

« 8. It is desirable in future in the case
« of new railway construction or when
« modifying existing lines to provide
« large cesses to facilitate the use of
« mechanical equipment and generating
« plant.

« 9. The application of scientific me-
« thods of organisation of work with the
« use of mechanical plant or hand tools
« for renewal and maintenance of the per-
« manent way with particular reference to
« reducing the difficulty of the work and
« increasing the output of the personnel
« is to be recommended on account of
« the technical, economic and social
« advantages which this offers. »

b) *Recent improvements relating to reinforced concrete and pre-stressed concrete sleepers. Results obtained.*

« 1. The use of concrete sleepers is
« still limited.

« 2. From experience already gained as
« far as non pre-stressed concrete sleepers
« are concerned, it can be concluded that
« the behaviour of these sleepers on lines
« with important traffic is not satisfac-
« tory. For this reason certain systems
« limit their use to secondary lines with
« light traffic or to sidings.

« 3. The systems which utilise non pre-
« stressed concrete sleepers in general
« prefer the mixed type consisting of
« blocks joined by tie-bars.

« 4. Pre-stressed concrete sleepers offer
« greater resistance to impact effects than
« do ordinary reinforced concrete sleepers.
« Having regard to the results obtained
« in laboratory tests it can be hoped that
« they will give satisfactory results even
« on lines with important traffic, and
« especially if they are fitted with elastic
« fastenings.

« 5. Improvements seem to be neces-
« sary in order that reinforced concrete
« sleepers may be used on electrified lines
« and on lines with track circuits.

« 6. The method of fastening the rail to
« the sleeper is very important as far as
« the protection of the latter is concerned.
« The types of fastening which have
« proved the most satisfactory are :
« — indirect fastening with separate fas-
« ening of the rail and the bearing plate,
« without any intermediate elastic plate;
« — a completely elastic fastening (elastic
« clip and non-rigid packing).

« 7. If the use of reinforced concrete
« sleepers is to prove economical compa-
« red with that of wood and steel sleepers
« their manufacturing costs must be
« greatly reduced below the present level.

« 8. It would be of interest to carry out
« tests regarding the use of long welded
« rails in conjunction with that of rein-
« forced concrete sleepers in order to
« determine the technical possibilities of
« this method and its financial repercus-
« sions. »

c) *Recovery and strengthening of metal bridges that have reached the theoretical limit of safety.*

« 1. The renewal or strengthening of a
« metal railway bridge is generally neces-
« sitated on account of increased rolling
« loads and also by increased speeds or
« on account of reduced sections due to
« corrosion.

« 2. The strength of existing metal
« bridges is determined by theoretical
« calculations often checked by direct
« measurement of the stress on the
« bridge in question.
« The permissible stresses for existing
« structures are generally appreciably
« higher than those allowed for the con-
« struction of new work.

« 3. Each case of a bridge which has
« reached its limit of safety should be
« examined separately in its own proper
« place in order to determine the best
« technical and economical solution and
« the relative merits of strengthening as
« opposed to renewal should be examined.

« 4. The re-use of complete bridges
« from other lines when circumstances
« allow can be an interesting and economic
« procedure.

« 5. Strengthening by welding is an
« economic and practical procedure which
« is being adopted to an increasing ex-
« tent : it requires to be carefully carried
« out.
« The use of this method for old iron
« bridges requires a procedure adapted
« to the nature of the metal.

« 6. The strengthening of sections by « rivetting is still the method most gene- « rally used, but it cannot be recommend- « ed when the addition of supplementary « parts involves removing the rivets from « important fastenings.

« 7. The strengthening of bridges by « the addition of new framework, con- « struction of new supports, etc., without « modifying the old structure is recom- « mended.

« 8. Strengthening certain bridges by « reinforced concrete has given good « results. In recent applications of this « method simple exterior shapes have « been preferred and effective bonding « between steel and reinforced concrete « has been sought. »

2nd SECTION : LOCOMOTIVE AND ROLLING STOCK.

QUESTION II.

Electric locomotives for fast trains (75 m.p.h. and over).

Discussion of adopted and projected types.

1. **Arrangement of the axles.**
2. **Type of axle drive :**
   a) **motor suspended from the nose;**
   b) **flexible transmission.**
3. **Electric motor characteristics.**
4. **Braking.**

Summaries.

« 1. Concrete achievements show that « the construction of locomotives for « speeds running up to 160-180 km./h. « (100-112 m.p.h.) is possible. The prin- « cipal and most severe technical restric- « tions to the adoption, for normal « services, of speeds of this kind arise « from the layout of the lines, super- « elevation, and signalling requirements, « in conjunction with the minimum num- « ber of stops.

« 2. The design of the mechanical « portion of locomotives, where the ar- « rangement of axles is concerned, is « influenced by the prevailing conditions « on the different railway systems; which « explains certain important differences « which are noticed between the U. S. A. « and European methods of construction. « American construction tended towards « an arrangement of axles with a guiding « bogie and articulated frames. The « present tendency, however, in all coun- « tries tends towards locomotives fitted « with motor bogies on 2 or 3 axles and, « in general, with total adhesive weight.

« 3. The numerous systems of axle- « drive in use appear to be still in a state « of evolution, although constructors « appear to prefer solutions allowing a « certain liberty of movement between « the ensemble of the toothed wheels in « mesh and the axle or else the motor- « shaft.

« The nose suspended motor is still in « use and it is even being adopted in « certain countries for new stock now « being designed.

« 4. There is no special features as « regards the traction motors used on « fast locomotives.

« 5. The problem of braking still exists « in spite of progress made for a better « use of adhesion and friction and ap- « pears as the most difficult to solve in « view of the great power to be dissi- « pated. »

3rd SECTION : WORKING.

QUESTION III.

Transport of miscellaneous goods.

Concentration in a certain number of selected centres (stations) of miscellaneous traffic, transport by rail between centre-stations, by road or rail between the originating point and the nearest centre-station, and also to the last centre-station near the destination.

Interest of the scheme for the conveyance of goods traffic.

Organisation of the station-centres and of the collection and delivery services.

Financial results of the scheme.

Summaries.

« 1. Miscellaneous goods are generally « conveyed by through wagons and by « pick-up and distributing wagons, or « separate pick-up and distributing wa- « gons or by wagons used for all purposes.

« From a point of view of output as « well as economy, and in order to im- « prove the service, it is advantageous to « adopt an organisation which allows of « an increase in the average load carried « by wagons, and of a reduction in the « labour involved and in an acceleration « of movement. This result is achieved « when the number of through wagons « reaches a maximum; due consideration « being given to each wagon carrying a « minimum load, the latter depending on « the operating conditions on each Rail- « way.

« 2. The method to be followed by « those Railways where the density of « traffic warrants it, is to send miscella- « neous goods by rail between a certain « number of centre-stations, carefully « selected, whence they can be distributed « or towards which they can be concen- « trated by means of road and/or railway « services. One can assume, as a result « of the improvement of the service, a « consolidation or even an increase in « traffic and a reduction in the number « of claims for delays, losses, thefts and « damages.

« 3. When dividing up miscellaneous « traffic between rail and road transport, « account should be taken of the extra « expense incurred by road transport, the « economies which Railways can effec- « tively realise in rail transport and the « advantages peculiar to each mode of « transport.

« This division may prove a stage in « the co-ordination of rail and road.

« 4. Traffic dealt with by road services, « in substitution of, or complementary to « railway services, should preferably be « carried under the responsibility of the « railways, the latter making direct con- « tact with the traders.

« These road services should be subject « to the same obligations as those of the « public railway services.

« 5. Goods should be sent by through « wagons as far as possible from centre « to centre, or, if necessary, through a « minimum number of transhipping sta- « tions.

« The most expeditious and economic « methods of handling and sorting miscel- « laneous traffic at such centres deserves « special study, particularly the methods « of mechanised handling.

« 6. As night work has a favourable « effect on the transit time, it should be « instituted at the stations where the « necessity arises.

« 7. In the countries where such sys- « tems exist the part played by grouping « agents and agents is of advantage to the « extent that these intermediaries facili- « tate the concentration of traffic and « render to the traders services which the « railway is not normally in a position « to provide.

« The rating benefits wherever granted « to grouping agents should depend on an « undertaking by the latter to limit their « activities to the sector given them by « the railway and to deliver to the railway « all the traffic it can deal with.

« The fees paid to agents are made « good by the possibility of making « economies in the staff, either by closing « certain stations to parcels traffic, or « reducing the number of employees.

« 8. If there is not much traffic on « certain sections, the number of road « services can be limited to 3 or 2 a week « (thereby increasing the transit times), « or other existing methods of transport « made use of which can carry out the « terminal transport without additional « cost (vans of passenger or goods trains, « light railway services, buses, etc.).

« 9. Where the system of centre-sta- « tions is in use the total price charged « should be as far as possible independent « of the method of transport utilised.

« 10. The organisation of the station- « centre system will allow, in a number « of cases, of economical development of « delivery and collection services.

« It is desirable that the rates for these « services should take into account the « weight of the consignment and the « distance it is carried. »

NEW BOOKS AND PUBLICATIONS.

[625 .28 (44)]

Locomotives des Chemins de fer français, à vapeur, électriques, diesel-électriques et autorails *(Locomotives of the French Railways; steam, electric, diesel-electric and railcars)*, No. 1, October 1947. — With a foreword by Paul LEGREGEOIS, Divisional Inspector, Locomotive Design; S.N.C.F. — Album ($6 \times 9^1/_2$ in.) of 72 pages, copiously illustrated. — 1947, Paris (XVII[e]), Editions P.P.C., 39, boulevard Berthier. (Price : 250 French francs.)

This is a handsome volume, published by « Editions P.P.C. » of Paris. It presents numerous examples of French motive power units, 17 types of steam locomotives, 6 electric locomotives or motive units, 2 diesel-electric locomotives and 4 railcars.

Steam locomotives are shown by photographs, with a dimensioned outline diagram and a table of main characteristics alongside each illustration. These show the predominance of compounding; the increased boiler pressure, which remains at 20 kgr./cm² (284 lbs. per sq. inch) in all the latest engines; the wide fireboxes, which are general on all powerful locomotives with one or two rear carrying axles; and the increased number of driving axles, reaching 4 for passenger engines and 5 or 6 for freight engines. Two locomotives are provided with mechanical stokers and one is, apart from the driving wheels, completely streamlined. There is a 3-cylinder compound — the 242A, and a 6-cylinder compound — 160A. The latter rises steam superheated and re-heated in steam jackets.

Amongst the performances noted may be mentioned the 120 km./h. (75 m.p.h.) with 800 tons achieved by a S.O. (S.W. Region) *Pacific* on level line, and 120 km./h. on a 8 °/₀₀ gradient by a Nord *Baltic*.

Of the four electric locomotives, only the 2D2 of the S.O. (S.W. Region) has all axles fitted in one underframe; the remainder have articulated frames. Only one of the four has rod transmission. The highest adhesive weight (107 400 kgr. [236 800 lbs.]) and the greatest power (3800 H.P.), are achieved by the S.E. Region's 2CC2. This is capable of hauling 700 tons at 90 km./h. (56 m.p.h.) on an 11 °/₀₀ gradient, and 350 tons at 45 km./h. (28 m.p.h.) on a 30 °/₀₀ gradient.

The two electric railcars are on vastly different scales; one is 1 500 H.P., with a weight of 78 000 kgr. (171 960 lbs.) and 132 seats, the other 750 H.P., 73 000 kgr. (160 940 lbs.) and 68 seats.

The same applies to the two diesel-electric locomotives. A traffic engine for fast services has 2 Sulzer, 12-cylinder motors, totalling 3 500 H.P., with 2 generators, electrical transmission and 6 motors. The total weight is 230 700 kgr. (508 600 lbs.). The second is a shunting locomotive of 300 H.P. and a total weight of 68 500 kgr. (151 000 lbs.).

There is a great variety in the railcars. The motors are both diesel and petrol types. Power varies between 266 and 810 H.P. The smallest caters for 48 passengers whilst the largest provides for 42 first class and 96 third class. The latter has electrical transmission, the other three mechanical transmission. The Michelin car is mounted on pneumatic tyres. The speed record of 150 km./h. (93 m.p.h.) is shared by the Franco-Belge triplet cars and the Bugatti cars.

These brief remarks show clearly that the collection is well representative of modern stock, the result of advanced research and of fruitful co-operation between the French Railways and the national industry.

E. M.

[621 .13 (09 .3 (42)]

LIVESEY (H. F. F.). — **The locomotives of the L. N. W. R.** — One volume ($5\,^1/_2 \times 9$ in.) of 100 pages with numerous photographs. — 1948, London, The Railway Publishing Co. Ltd, 33, Tothill Street, Westminster, S. W. 1. (Price : 10 s. net.)

Before the amalgamation of the British railways into four main Companies, there were in the country a large number of separate undertakings. Competition between them was not always free from disadvantage and this provided one argument for reform. On the other hand, rivalry encouraged improvement and, helped by the British sporting instinct, consistent improvement in locomotive construction was maintained, with some notable resulting performances.

It was the admiration of the Author for the trains in which, as a boy, he regularly travelled North, which led him to collect the framework of this book. The L.N.W.R., absorbed in 1923 into the L.M.S., was itself formed by the amalgamation of several lines, amongst them the Liverpool and Manchester. As mentioned by Mr. Chas. E. Lee in the preface, this was early in the history of British railways. and was a link in the first main line system. The work by Mr. Livesey is, therefore, intimately connected with the history of the first railways. His notes commence in 1846, when the L.N.W.R. was formed.

All the locomotives introduced since then are recalled, with their characteristics, class, name (the practice of which has been carried on for a very long time) and the success which they have achieved.

In this succession of different designs can be seen the evolution of the locomotives : the boiler transformations, the continuous increase in power and boiler pressure, the appearance of superheating, the struggle between simple and double expansion, valve gears ending in the supremacy of Walschaerts, and increases in numbers of axles. The single drivers, which persisted obstinately, had eventually to give way to the 4-6-0 and finally to the *Pacific*. The author quotes appropriately well-known names, mostly now no longer with us, all of whom have contributed to locomotive progress.

The book concludes with a fine collection of photographs, some of the locomotives being very early ones. These provide a further source for appreciating the simplicity of line which distinguishes British construction.

E. M.

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[694]

CORKHILL (Thomas), MI.Struct.E., F.B.I.C.C. — **A Glossary of Wood.** — One volume ($5\,^1/_3 \times 7\,^1/_2$ inches) of 656 pages, copiously illustrated. — 1948, London, published by the Editor of the review « Wood », 33, Tothill Street, Westminter, S. W. 1.

This glossary has been prepared for the benefit of all those interested in the timber industry, whether as owners of plantations or as timber merchants, or in the preparation of wood for its many uses. Those in charge of plant, technicians and draughtsmen need to be in possession of a large sum of information which they will find here in alphabetical order.

The work contains some 10 000 articles more than 1 000 of which are

accompanied by very clear drawings stressing the explanations given. The terms cover the different kinds of timber, their conservation, their preparation for diverse uses, the tools used, the different uses both in carpentry and outside use as for decoration and furniture. Many drawings of completed assembles are amongst the most interesting illustrations.

The matter of this considerable work was first published in periodical notes in the publication *Wood*, the illustrated monthly review devoted entirely to the timber industry. The interest these aroused decided their author to collect them into book form. This gave him the chance to amplify them somewhat and extend his list, whilst making a general revision of the articles retained and definitions given.

It is only necessary to read a few pages to appreciate the care with which the author has edited each note. One difficulty which he had to overcome is the frequent use of several different names for one and same wood. He frequently had to have recourse to the scientific botanical name. He has also taken the work of the *Forest Product Research Station* and the *British Standard Institution* as a basis.

There is no doubt but that in meeting the wishes of subscribers to *Wood*, the author has supplied them with a work of reference which will greatly facilitate their professional work.

E. M.

[**656** .254]

VERSTEGEN (J. H.), Head of the Signals Department of the Netherlands Railways. — **Automatische beveiliging van niet-afgesloten overwegen** *(Automatic protection of level crossings, where there are no keepers)*. — One volume ($5^1/_2 \times 7^7/_8$ in.) of 66 pages with figures and tables *A* to *P*. — 1947, Utrecht, published by N. V. Nederlandsche Spoorwegen.

The question of level crossings underwent an appreciable change in Holland owing to a modification in the laws in 1922. The new regulations made it possible to do away with keepers at crossings meeting certain given conditions. In a few years keepers were suppressed at 1 400 out of the 2 600 level crossings on the main lines (hoofdspoorwegen). After several serious accidents had occurred, a national commission was set up to hold an enquiry into the safety of crossings without keepers on the large and small railways. They also had to investigate if and when equipment giving a warning of the approach of a train was possible and desirable. This gave rise to the problem of automatic protection.

The author, who has given special attention to this aspect of safety on the railway, has collected together in this volume various notes published by him in the reviews *De Ingenieur* and *Spoor- en Tramwegen*.

The principles of the solution adopted by the N. S. (Nederlandsche Spoorwegen) are dealt with in Chapter III. Winking lights are used : a white light when the crossing is clear, a red light when a train is approaching. In addition, there is a steady orange light in case anything goes wrong. The red lights wink faster than the white lights. A bell rings at the same time as the red light shows. There are also other very interesting arrangements.

A trial installation was made at Steenwijk. Entirely electric, this presented no snags for the Dutch engineers who were experienced in the use of the automatic block and many applications of insulated rails and insulated track. It was a matter of finding out if the public would profit by this improved installation. It is curious to read in this connection the result of the

counts made as well as those of a similar enquiry carried out in the United States. At the time in question (1943) some 40 installations were in service and an equal number planned.

At Steenwijk, the matter was simpler: a double track on a non-electrified line. All the explanations given regarding electric circuits, the working and the equipment are given in Chapter IV. The system is based on track circuits with rest period current, which flows as long as there is no train on the track.

On the electrified lines, special devices have to be used. These are described in detail, together with diagrams showing the electric circuits, in Chapter V. We will merely state that use is made of resistances through which the traction current can flow but which offer great resistance to the passage of the alternating current used for the signals.

In the case of single track lines, it was not found possible to use the plans for double track without modification. The author gives the solution adopted by the N. S., together with complete diagrams, a solution which, contrary to that adopted elsewhere, keeps the signal at danger when there is a train standing on the crossing itself.

In the final chapter, which forms an appendix, will be found a description of the light signals and of various parts of the equipment and the rules concerning working, fitting and maintenance.

This book will be of interest to specialists by giving them full details of an original solution which the inventors intended to be as satisfactory as possible for the road users.

E. M.

[**385.** (09 (44)]

LARTILLEUX (Henri), former pupil of the Polytechnique, First Engineer of the S.N.C.F. — **Le Réseau National des Chemins de Fer Français.** *Histoire et Organisation. (The French National Railways. — Their history and organisation).* One volume ($6^{5}/_{16}'' \times 9^{27}/_{32}''$) of 100 pages, copiously illustrated. — 1948, Paris, Editions P.P.C., 39, boulevard Berthier. (Price : 275 French francs.)

This work is both a monograph and a short history of the French National Railways. As it merely sketches in the main outline and stresses the striking facts, it is as pleasant to read as it is instructive.

The French National Railways had hardly come into being when war broke out in 1939. This means to say that events prevented the early application of the projected organisation methods. In addition, they had to go prudently, basing themselves on the solid foundations bequeathed them by the former railways, who had behind them eighty years experience.

Logically, the author begins by giving the origins, formation and development of each of the great railways, together with the characteristics of the districts served and the nature of the traffic. The number of these railways had been reduced to five by the amalgamation of the P. O. and Midi, and the lines of the Ouest and State (Etat) systems. It was the financial difficulties in which they found themselves, which had nothing to do with their management, which led the State to put the operation of the entire railway system into the hands of a National Company, an autonomous Company but under its direct control. The author recalls the beginning of the S. N. C. F., the trials of the war and occupation, the disastrous situation at the liberation, and the period of reconstruction. At the same time the train services were being restored as rapidly as possible,

the S. N. C. F. was preoccupied with the final reconstruction and modernisation of the system.

As regards the present organisation, the author gives very substantial details and characterises the tendencies. Unlike the old organisation which was extremely centralised, the current trend is towards progressive decentralisation, the first steps having already been taken by the setting up of a new regional management, the « Regions » originally decided upon having simply replaced the management of the railway.

In the chapter devoted to the *Staff,* besides other headings, will be found those dealing with the recruiting and training of the staff, a very important question for an industry employing large numbers like the railway, with such diversity of activities.

In the following pages, the technical aspects are covered, using the expression in its widest sense, i.e. the whole equipment of the railway both fixed and mobile, together with the way it is used. The layout and profile of the lines, the constitution of the superstructure (with the proposals to increase the axle loads), signalling (with the projects to extend the automatic block using light signals), the extension of electric traction, the locomotives and rolling stock are all subjects which enable the author to show how the S. N. C. F. proposes to modernise its system. New methods will also be applied in the organisation of the passenger traffic and the transport of goods, both on the railway itself and its extensions. This brings us to the supplementary activities of the railway. The measures taken to co-ordinate transport and the intervention of the railway in road transport must be included in the methods of meeting the requirements of clients whilst safeguarding the interests of the railway.

The author ends on an optimistic note. Both in the chapter devoted to the future programme and in his general conclusions, he considers that the new technical strength and the orientation given to the Operating Services will make the reconstructed French railways a national tool well fitted to the very varied requirements of the national economy.

E. M.

[621 .132.8]

DE GREGORIO (Camillo), Professor. Engineer. — **Caratteristiche fluodinamiche e « capacita di lavoro » delle locomotive ad accumulatore termico. Prospettive per la loro carica mediante energia electrica.** *(Evolution of the dynamic characteristics and working capacity of locomotives with thermic accumulators. Possibilities of putting them under load by means of electric power).* Reprint from the review « Ingegneria ferroviaria », No. 7-8, July-August 1948. — One brochure ($8^3/_4 \times 11^1/_2$ inches) of 12 pages, with tables and figures.

This is a study of a proposed locomotive without firebox. The source of heat is electricity. The fixed electric heating installation consists of a water accumulator in which the water is heated to a very high temperature. This accumulator provides the steam for the boiler when the locomotive is refuelled.

To obtain satisfactory output and a good working capacity, the initial pressure in the boiler has to be very high. The value reached in the dome by means of a reducing valve which lowers the pressure to the ordinary working value is superheated by passing through a coil in the water.

As the basis for his research the author gives a certain number of original diagrams which he has prepared according to the known properties of steam :

Heat contained in 1 kgr. and in 1 m^3 of steam at different saturation points.

Kilograms of saturated steam to be

obtained per m^3 of water stored as a function of the pressure under load for various values of the pressure used.

Number of litres of water to be stored to produce 1 kgr. of saturated steam for the pressures given for load and utilisation.

Number of kilograms of steam to be obtained per m^3 of water stored for the given pressures of load and utilisation, in terms of the difference in pressure.

A study of the conditions under which steam is superheated and the circumstances which affect the consumption of steam is given by means of other original diagrams.

One interesting idea on which the author insists is what he calls the working capacity of the locomotive. He expresses this in units (C.L.) each of which equals the product of 1 km. run with a drawbar pull equal to 1 000 kgr. This is related to the current working values, the horse power hour and the kilowatt hour by means of equations :

1 unit C.L. = 3.7 H.P. hours = 2.72 kWh.

Diagrams and numerical tables show the variation in the working capacity in terms of the weight of the thermic accumulator locomotive and for various values of load and utilisation.

The field of use of the proposed locomotive would seem to be the shunting services in the stations of an electrified system where all the secondary lines are not yet or will not be equipped electrically.

One advantage claimed is the little time taken for recharging operations and consequently the high coefficient of use of the locomotive. From this point of view the electric accumulator locomotive is inferior.

The author gives some interesting figures in this connection. He also extends his comparison to the questions of the weight, cost and output.

His conclusion is that the electrically heated thermic accumulator will be very useful in the transport field, and not only on the railway. It may be added that part of the value of this method is due to the author's hypothesis that the source of electricity is hydraulic.

E. M.

[656 .222.4]

GUTH (Hans), Zurich. **Die Schnelligkeit im Eisenbahnpersonenverkehr.** *(Rapidity in the transport of passengers by rail).* — A discussion presented to the Faculty of Judicial and Administrative Sciences of Zurich University to obtain the degree of Doctor of Economic Science. — One volume (6 · 9 inches) of XXVIII + 294 pages. — 1948, Zurich, Genossenschaftsdruckerei.

The expression used to translate the title seems to us to translate both the wording of the title and the object of the thesis better than the word speed. It is question in fact not only of the actual speed during the railway journey, but also and above all the general promptitude with which passengers are transported from their homes to their actual destination. That is to say the question is examined very thoroughly under every aspect.

Various elements come into account to make up the total duration of a journey. The author mentions them and briefly analyses them in his introduction. There is the journey to the station, which depends on the situation of the station relatively to the town, and the operations which take place before the train is boarded, which depend on the arrangement of the offices and the organisation of the service. On arrival similar problems arise.

As part of the time taken to make a journey, we must also consider if needs be an arrival too early in the morning or

a late departure at night. From this point of view, the timetables should be adapted to suit the convenience of passengers, and the more train services there are, the more likely is this to be achieved.

Everyone knows the desastrous effect of intermediate stops on the journey time between two given places. A few striking figures show how little effective is an increase in the speed, even a considerable one, if it is not possible to reform the number of stops and their length at the same time. This raises the question of connections and the specialisation of trains which will be discussed when dealing with the structure of the timetables.

In countries where the configuration of the country and the density of the system are such that part of the people only have indirect access to the fast lines, there is a serious loss of time due to complicated timetables. The timetables should be arranged to reduce this as far as possible.

Finally, as the author also has in view a comparison with other methods of transport, it should be noted here on behalf of the railway that it is possible, owing to the comfort and various facilities offered, to discount much of the time passed in a train as being time lost.

To complete his introduction, the author includes reflections on the interest of speeding up communications showing the effects of a policy of continuous improvement on the economic and social life of the country.

These summary indications are sufficient to enable readers to judge of the way in which the various successive stages in the evolution of railway transport are investigated and the circumstances which have influenced such ideas and facts.

First of all there is the period before the automobile and airplane came on the scene. In spite of the considerable improvement achieved by the substitution of the railway for the old methods of travelling, the railway undertakings have never ceased to look for methods of keeping the train services on a level with the needs of the public.

In making a comparison as regards speed with the automobile and the airplane, the author does not ignore any of points for and against these two competitors. As regards the automobile, he ends by defining two zones which according to the journey time will fall one to the road, the other to the railway, leaving between the two a large contestable area, a sort of no man's land open to both competitors, without excluding the possibility of incursions into the opposite field. A thorough discussion taking into consideration all the elements concerned such as the length of the lines, the running speed (influenced by the stops made), the journey to and from the airport, the frequency of departures, the times of departure and arrival, the degree of regularity, the comfort, makes it possible to set approximate limits to the field of competition from the airplane. During the discussion, an allusion is made to the very high cost of air travel, and the financial assistance given by the public authorities, which directly or indirectly favours the airplane in its competition against the railway.

The author's opinions are based on facts and ideas collected from many sources. These are indicated in a separate list and in many foot notes which sometimes contain remarkable quotations. These facts have enabled him to discuss, a propos the increases in speed realised between the two wars, the relations between the speed and the cost. According to him, the compensating economies which incline the weights to the other side have frequently been neglected.

The discussion on the surcharges made for the express and fast trains is also well developed, as well as the low charges on certain trains. The result is that the idea of a maximum rate has often been lost of sight (the amount the client is disposed to pay), as well as the repercussions of such a policy on the user of the trains.

This question is closely linked up with that of the admission of cheap ticket holders to fast trains and the gradation

of supplements if need be. Amongst the many arguments discussed, mention must be made of those concerning the commercial spirit and those based on the complications already caused by supplementary fares, and the worse ones of having variable supplements.

A first appendix gives comparative costs for the train-kilometre with a locomotive and railcars of various sizes in several countries of Europe and America. A second shows the amount of the speed supplements in those countries where they are charged side by side with the normal fares.

E. M.

[**625** .143 (0]

DE PATER (A. D.). Mechanical Engineer. **Inleidend onderzoek naar het dinamisch gedrag van spoorstaven.** *(Introduction to a study of the dynamic behaviour of rails on the track).* Thesis submitted for the degree of Doctor of technical science. — One volume ($6^1/_4 \times 9^1/_2$in.) of 154 pages and 50 figures. — 1948, Delft, Uitgeverij Waltman.

The study of the bending of the track rails due to the rolling stresses in the axles is very complicated. Even if one restricts the problem to purely static considerations, it is still most difficult to solve unless one has recourse to simplifying assumptions. These latter cannot be considered as even approximately valid until it has been possible to verify by experiment, the results indicated by theory. Against this, carefully conducted tests may provide valuable indications for directing further research.

It was thus that at the commencement of this century, the Belgian engineer Flamache, as a result of tests undertaken by means of his apparatus known as the « deflectograph », was able, by analysing the diagrams obtained, to obtain confirmation of his treatment of the rail as a continuous elastic component supported along the whole of its length. Moreover he found that the flexure of the rail was to a very large extent independent of the spacing of the sleepers. His calculations led him to formulae, which in conditions taken from existing tracks, gave a maximum bending moment fairly close to the upper limit 0.250 Pl corresponding to that of a bar simply supported.

This engineer also investigated the dynamic aspect of the problem and came to the conclusion that the ordinary laws of bending are not applicable to the rails subjected to the action of fast trains.

Rail vibrations are likewise the subject of M. de Pater's investigation. It is a very lengthy essay which testifies to its author's regard for the exactitude of his mathematical reasoning. One notices too, the author's constant concern for the mutual support which theory and experience are bound to accord to each other.

Certainly the author himself has been obliged to admit the justice of certain assumptions or rather certain ideal concepts. He justifies them by facts ascertained in course of trials and by logical deductions and he shows how they can be linked up with the actual stresses arising.

The starting point, is the fundamental equation in general mechanics which determines the equality of the forces acting on an element of the rail and the product of its mass mutiplied by its acceleration. Among the forces concerned are the reactions at the points of support. These are represented by a continuous function of the abscissae and of time. The same applies to the vertical loads. In order to take into account the masses that take part in the vibration of the rails (partly in the superstructure and partly in the substructure) the author admits that one may increase the mass per unit of length by

an amount to be determined for each case and at the same time adding to the equation a term related to the damping effect. This latter is proportional to and of opposite sign to the speed. Finally a fourth term will represent the differential of the shearing stress. In this way we are led to a linear differential equation of the 4th order.

The integration works out equally for the free vibrations (in the absence of any external applied load) as for the forced vibrations. The former are governed by the sum of the two products of two functions, one depending on the abscissa while the other is solely related to the time. In the case of the forced vibrations, we have to deal with an indefinite series, of which the terms are products of the same functions of the abscissae and of new functions of the time.

After explaining the general method of the calculation, the author treats numerically, the case of a continuous rail carrying a single load and that of a rail provided with an elastic articulation (i.e. fished joints).

In the next chapter, the functions are calculated defining the free vibrations for different cases: rail of constant rigidity and of constant mass per unit of length, rail formed of two similar pieces connected together by an elastic hinge, rail having its mass concentrated at the centre.

The forced vibrations are studied under the supposition that they are due to a rolling load at constant speed, but variable periodically in time. They present the special case of a fixed load but variable periodically in time and that of a moving load but of constant weight.

The differential equation at the start being linear, it is easy to calculate the bending of the rail due to any periodic force by developing it according to a Fourier series. This method allows the author to consider the case of the track travelled over at constant speed by a railway vehicle, noting firstly the masses carried eccentrically by certain wheels and then the lack of circularity of the wheels.

In the last chapter, the author describes the arrangement as he thought it out and the tests which he undertook, with the object of verifying the value of the hypotheses put forward.

Finally in a concise analysis, he recalls the work of his precursors, with which he was familiar, indicating the point of view which they took up, the theories which they accepted, the cases which they had taken into consideration and how in his opinion it would be possible to complete their studies.

Although he gives it the unpretentions title: « Introduction to a study of the behaviour of rails », this thesis appears to be an important contribution to a study of the construction of the superstructure of the permanent way and one may consider his object as achieved, namely to see his thesis serve as a basis for further experimental research. The theories developed are moreover susceptible to further applications, such as may arise in exemples quoted in respect to the effect of unbalanced counterweights and with reference to contributary vibrations of the masses in suspension.

E. M.

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